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CONTENTS

Page 3: DESIGN FEATURES OF THE 60,000 kW TURBO-GENERATING SETS AT THE J. CLARK KEITH POWER STATION, WINDSOR, ONTARIO, CANADA, *by A. C. Hirst, B.Sc. (Eng.), M.I.Mech.E., M.I.E.E., M.Inst.F., Sales Manager, Steam Turbine Division.*

10: THE NAPIER SPINNING RIG, *by P. J. Wallace, Chief Designer, Test Plant Department, D. Napier & Son Limited.*

21: TEKAPO HYDRO-ELECTRIC PLANT, *by H. L. Bazalgette, A.C.G.I., M.I.E.E., A.M.Inst.C.E.*

35: SUPERVISORY AIDS IN HIGH PRESSURE HIGH TEMPERATURE STEAM TURBINE OPERATION, *by L. J. Higgins, Instrument Engineering Department, and P. F. Carson, Outside Erection Department.*

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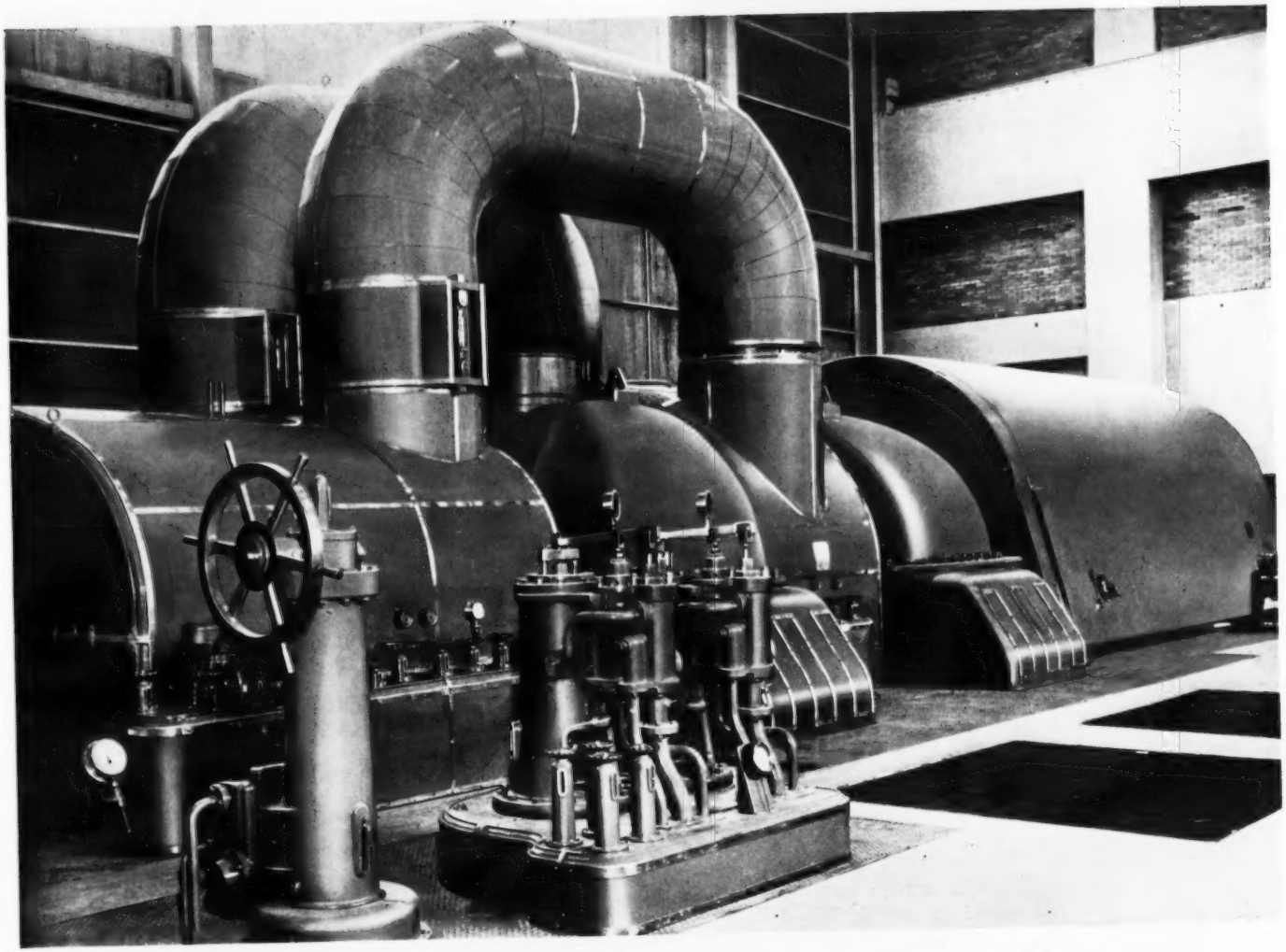


One of the 60,000 kW turbo-generating sets in the J. Clark Keith power station at Windsor of the Hydro-Electric Power Commission of Ontario, Canada

Design Features of the 60,000 kW Turbo-Generating Sets at the J. Clark Keith Power Station, Windsor, Ontario, Canada

By A. C. HIRST, B.Sc. (Eng.), M.I.Mech.E., M.I.E.E., M.Inst.F., Sales Manager, Steam Turbine Division.

THE J. CLARK KEITH power station at Windsor is one of two large thermal power stations commissioned in 1951 by the Hydro-Electric Power Commission of Ontario to supplement the power from the hydro-electric generating plants in the province.

The station was officially opened on the 16th of November, 1951, by Mr. Robert H. Saunders, Chairman of the Power Commission, and the first unit was started up by Mr. J. Clark Keith, General Manager of the Windsor Utilities Commission, in honour of whom the station was named.

In his address at the opening ceremony, Mr. Saunders paid tribute to the contractors who had made it possible to commission the first unit in little more than 28 months after the first ground survey had been completed.

The station is located on 110 acres of ground south-west of the city of Windsor, and is ideally situated with respect to both coal and water supply.

Negotiations for the building of the Windsor station were commenced in 1947 and it was decided to instal four 60,000 kW turbo-generating sets together with boilers and condensing plant. Orders were placed for the first two units early in 1948 and the remaining two have since been put in hand.

The contracts for the steam turbines and electrical generators were placed with The English Electric Company and these sections of the plant have been built at the works of the Group in England. The normal continuous maximum rating of each generating plant is 60,000 kW with a most economical rating at 48,000 kW and a ' capability rating ' of 66,000 kW.

STEAM TURBINE

The turbine is designed to run at 3,600 r.p.m., to operate with steam at 850 p.s.i.g., 900° F. (900 p.s.i.g., 915° F. maximum) and to exhaust to a vacuum of 28½" Hg. at 60,000 kW load.

It is a two-cylinder machine of the usual ' English Electric ' design comprising an impulse type high pressure cylinder and a double flow reaction type low pressure cylinder. This design offers the advantages of a robust rotor and ample running clearances at the high pressure end where the flow areas are small and pressure and temperature conditions more severe, together with the greater hydraulic efficiency of reaction blading at the low pressure end where areas are large and leakage losses proportionately smaller.

The steam supply is through two centre pressure stop valves to two cast steel steam chests arranged symmetrically about the turbine, each chest being provided with a combined main stop and emergency trip valve. A separate cast steel chest is welded on to each main steam chest and carries the main throttle valve and overload valve. The chests operate in parallel, each main throttle valve being connected to the turbine by two pipes and each overload valve by one pipe. Adequate flexibility in this piping combined with the symmetry of the arrangement ensures that pipe thrusts generally cancel out, thus relieving the turbine of any undue thrust loads from this source.

The high pressure rotor is machined out of a solid forging and the diaphragms in the H.P. cylinders are of built-up construction. The double flow low pressure rotor is a solid forging except for a separate disc shrunk and keyed on at each end carrying the last rows of blades.

The couplings throughout are of the multi-tooth flexible type with generous axial float and, as the H.P. turbine is provided with its own thrust

of the Hydro-Electric Power Commission of Ontario, Canada

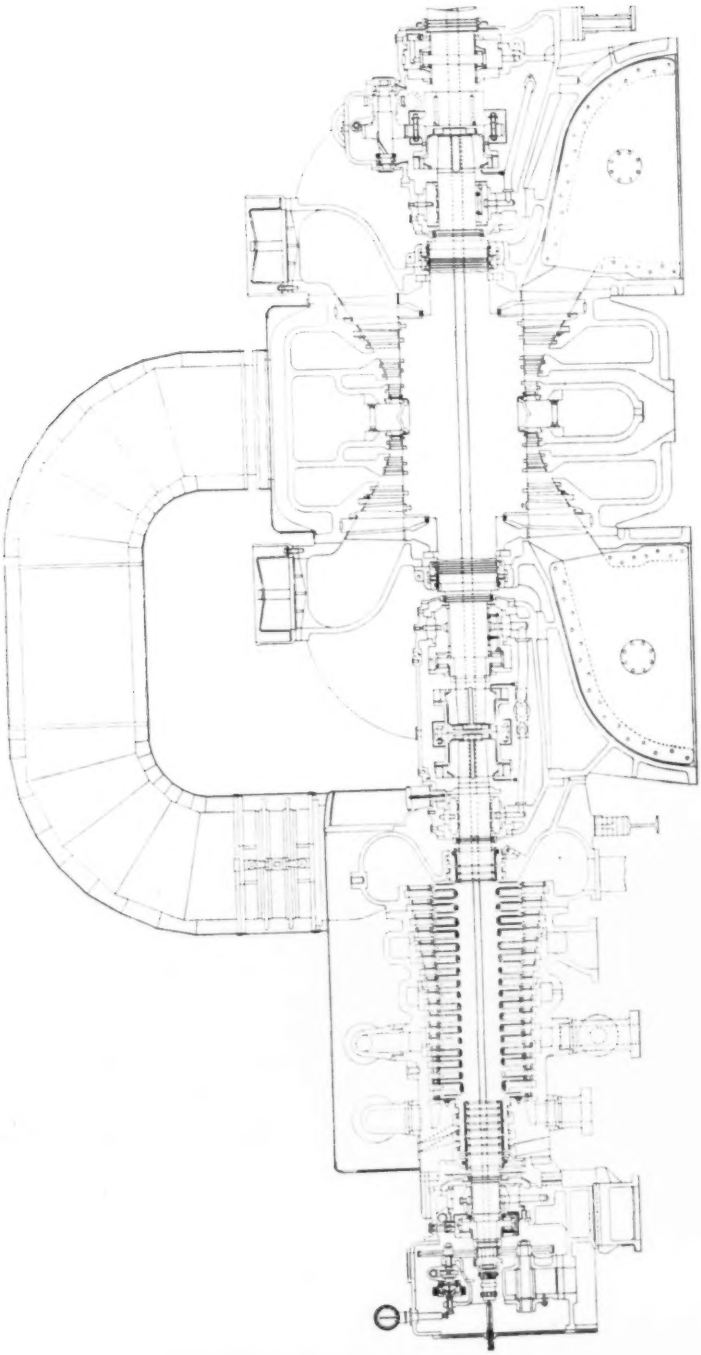


Fig. 1. Cross-section of turbine

block, differential expansions are confined to each individual cylinder. The H.P. turbine thrust block can be moved by a small amount in an axial direction while the turbine is in operation so that the effects of differential expansions during starting or unloading periods can be minimised. With the same object in view when starting, the flanges of the H.P. cylinder are provided with controllable steam heating.

Blading and Glands

The blading is of stainless iron or chromium plated 5% manganese nickel steel, depending on the stress conditions. It is found that manganese nickel steel is particularly suitable for withstanding the heating necessary when attaching lacing wires by silver soldering, without air hardening and the consequent risk of the formation of hair-line cracks.

All the moving blading is machined out of solid bar material and has integral roots. The blade fixings are of various types suited to the stresses imposed in service but mention may be made of the fixings of the long blades of the last row at each end of the L.P. turbine. These are of the ' side entry ' type in which each root is held in a separate groove in the disc head, the grooves being cut across the head in an approximately axial direction. The effect of this is that bending stresses are virtually eliminated, both in the blade root and in the disc head. The total resultant stress at this vital point is therefore much reduced.

The glands throughout the machine are of the labyrinth type, steam packed, and the gland system is so arranged that when the machine is on load the leakage steam from the various high pressure glands is conducted to lower pressure stages of the turbine where it performs useful work. The vacuum glands of the L.P. turbine are packed by steam taken from the H.P. cylinder at a moderate temperature. The system is also so arranged that during starting and stopping periods the glands are fed by steam at a temperature suited to the existing temperature of the glands at the time.

Governing

A hydraulic system of governing and load regulation is employed, the function of which is to convert the normal characteristic of the speed governor into a hydraulic oil pressure characteristic which is transmitted to secondary relays to control the opening of the steam valves and consequently the load on the machine. A small high-speed centrifugal governor is driven at 1,200 r.p.m. through gearing from the main shaft and operates a sleeve which is mechanically connected to an oil governing valve. Any movement of the sleeve transmits a corresponding

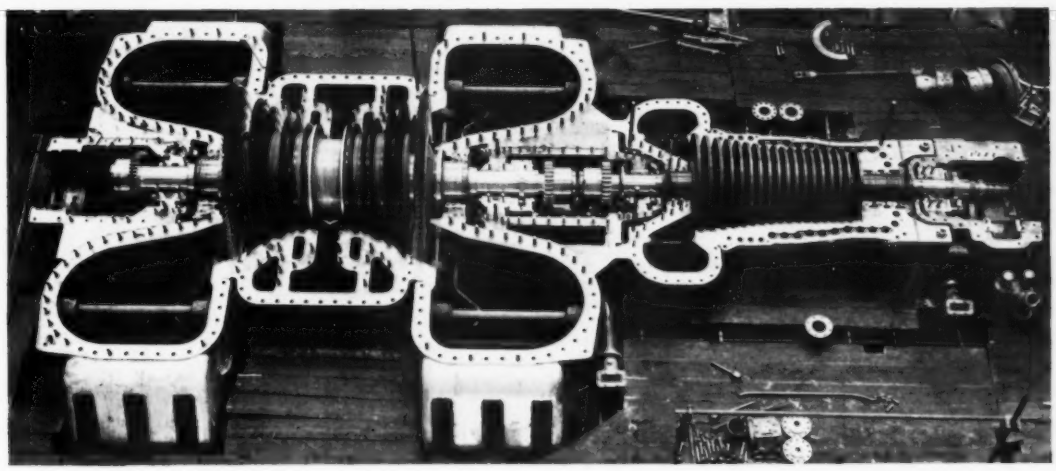


Fig. 2. Bottom half of turbine with shafts in position, at The English Electric Company's Works

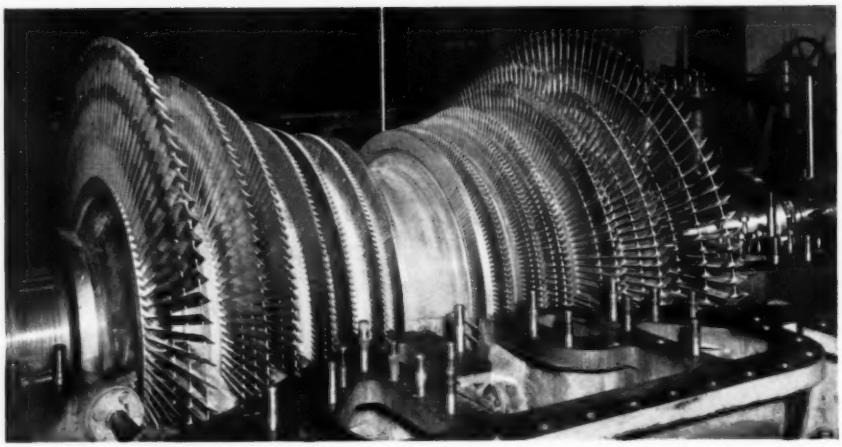


Fig. 3.
Low-pressure rotor in position in the cylinder

movement to the governing valve which, through the action of a bellows sleeve, increases or decreases the area of an oil leak-off port. This varies the oil quantity flowing to drain and consequently increases or decreases the oil pressure in a primary relay cylinder in relation to the characteristic of a spring opposing the oil load. The change in pressure is transmitted to secondary relays which in turn operate the steam valves through the medium of oil distribution valves and relay pistons.

To vary the speed or the load on the machine provision is made for hand operation or remote operation of the oil distribution valves.

The oil pressure characteristic is arranged to govern with low pressure at no load and high pressure at full load so that if loss of governing oil pressure occurs the steam valves will automatically close and shut off steam to the machine.

The emergency governor gear is actuated by a mechanical overspeed governor of the unbalanced ring type, by an electrical overspeed governor, by hand trip at the turbine or by an electrical trip operated from the control room. In each case both the emergency valves and the governing valves are closed so as to shut off completely the steam supply to the turbine. Once the trip has operated the mechanism remains in the tripped position until reset by hand.

Lubrication

The lubrication system comprises an oil tank and a main oil pump housed in the H.P. pedestal and driven from the main turbine shaft through helical gearing. In addition a steam-driven full duty auxiliary oil pump and an A.C. motor-driven lubricating oil pump are provided. As it is intended to uncouple the generator and to run it as a synchronous condenser under certain conditions of load on the system, two further lubricating oil pumps, one with A.C. drive and one with D.C. drive, are provided.

Oil is drawn from the tank and delivered through a non-return valve to the distribution box. This is fitted with a relief valve set at about 60 p.s.i.g. through which surplus oil is returned to the tank. Pressure oil is supplied to the governor relay gear from the distribution box.

Oil for lubricating purposes passes through a pressure reducing valve to the coolers and thence to the bearings at a pressure of about 15 p.s.i.g. Oil from the bearings passes through sight boxes to the bus return pipe, which conveys it back to the oil tank through strainers.

Oil alarm signals are transmitted to the main control board if the temperature of the return oil from any bearing should rise to an excessive value.

Barring Gear

Electrically driven barring gear is provided to reduce to a minimum any risk of the turbine rotors distorting due to uneven heating or cooling whilst the set is shut down. The gear turns the rotors at about 20 r.p.m. ; it is put into operation

by hand but is thrown out automatically when the turbine speed under steam exceeds that of the gear.

High pressure oil jacking is provided at the L.P. turbine and alternator bearings to ensure that each of these journals is supported by a continuous oil film before barring commences, until the journal speed is sufficient to maintain an oil film when supplied by the auxiliary oil pump. Damage to the bearings is thus avoided and the starting effort required from the barring motor is greatly reduced.

Supervisory and Protective Equipment

In order to obtain close control during starting up and shutting down periods, the set is equipped with supervisory instruments to give information regarding the condition and behaviour of various components, particularly in the H.P. turbine. The following instruments are included in this category :—

(i) An electrical speed indicator in addition to the usual mechanical tachometer, which indicates and records the speed from about 100 r.p.m. upwards. The speed is measured by calibrating the output from an induction tacho-generator driven from the main turbine shaft, and a record is provided in addition to the indicating instrument.

(ii) A shaft eccentricity indicator and recorder. This instrument is of great importance when starting up, particularly if the machine is still warm after a comparatively short period of shut-down. It indicates and records any eccentricity there may be at a suitable position on the H.P. rotor, thus showing if the rotor is bowed due to unequal cooling. This information is obtained whilst the rotor is revolving at low speed, before any harm would be done, and the operators are thereby warned not to increase speed until the shaft has become uniformly warmed and has thus straightened itself.

(iii) A shaft position or axial differential expansion indicator and recorder. This is provided to show the relative expansion between the H.P. casing and the rotor and is again of especial importance when starting up and shutting down the machine. Suitable action by the operators, dictated by the readings obtained from this instrument, enables axial clearances in the H.P. turbine to be kept at safe values even though the casing and rotor are expanding or contracting at different rates.

The last two instruments operate on the principle of introducing the distance to be measured as a variation in an air-gap controlling the reactance of an inductive circuit.

(iv) An H.P. pedestal expansion indicator. Pedestal expansion is indicated and recorded by measuring the relative motion between the steam end pedestal and the pedestal base. Any displacement of the pedestal with respect to the base initiates movement, through a lever arm, of an iron core sliding within two solenoids in such a way as to vary the relative amount of iron inside each coil. The relative values of the two inductances so formed determine the output of a bridge circuit, and expansion is indicated and recorded by calibrating the bridge output over the required range.

(v) Vibration indicators in the form of pick-up heads are mounted on the bearing pedestals just below the centre line of the turbine shaft in such a manner as to measure transverse vibration amplitude. The readings are indicated and recorded.

Vacuum Unloading and Trip Gear

Vacuum load-limiting gear is provided to unload the set gradually by action on the governor gear, commencing to unload when the vacuum has fallen to about 23″ and continuing until, at about 17″ of vacuum, all load has been removed.

Blade Washing

Arrangements are provided for supplying the turbine with dry saturated steam to wash away any deposits formed. Blade washing is carried out at no load and at a considerably reduced speed.

ALTERNATOR

The alternator is hydrogen-cooled and generates at 13.8 kV, 3-phase, 60 cycles. The rated output is 60 MW at 0.85 p.f.

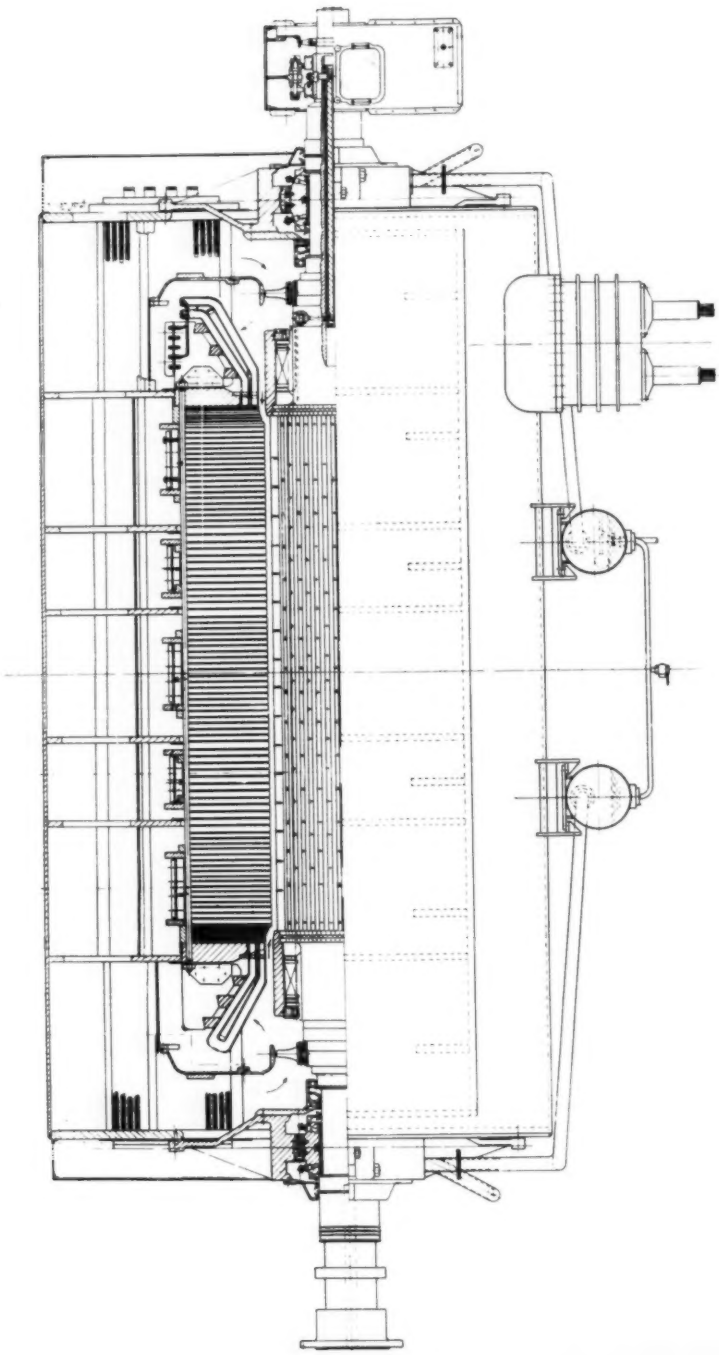


Fig. 4. Cross-section of the hydrogen-cooled alternator

Stator

The stator frame is a fabricated structure, ribbed and stayed internally to ensure rigidity. The bearings are supported by separate cast steel brackets bolted to the end-plates of the frame. Precautions are taken in the operation of the machine to prevent the formation of an explosive mixture of air and hydrogen, but to provide for some unforeseen condition the stator frame is designed to be "explosion safe," that is, it is built to withstand without failure any pressure set up by an explosion.

The stator core is built up from segmental stampings carried in a skeleton frame inside the main stator frame by flexible leaf springs. Any 120-cycle vibration produced by magnetic attraction between the poles of the rotor and the stator core is thus largely prevented from being transmitted to the main structure and the foundations, whilst at the same time the core is rigidly restrained against the load and short-circuit torques.

The alternator is entirely self-contained in that the two gas coolers are mounted axially in the upper part of the outer frame and the gas is circulated by means of axial flow fans. There are two fans, with blades of aerofoil section, one fan being mounted at each end of the rotor. This construction eliminates all external ducting and thus greatly simplifies maintaining a gas-tight enclosure.

Rotor

The rotor is of conventional design and is coupled to the turbine through a multi-tooth flexible coupling. It is machined from a single forging of carbon steel with additions of manganese, nickel and vanadium. Ventilation channels are provided beneath the slots and down the centres of the teeth. As the result of cutting the slots to contain the field winding, the inertia of the rotor is different across the two major axes, so that the static deflection at the centre of a long rotor such as this is appreciably greater when the pole axis is horizontal than when it is vertical. This effect gives rise to a vibration of the rotor at twice the running frequency, and in order to eliminate this as far as possible transverse grooves are machined in the pole centres.

The field winding is formed from semi-hard silver-bearing copper in order to prevent coil distortion through differential thermal expansion. The turns are taped throughout their length with glass-backed mica tape, which completely eliminates the risk of short circuits developing between turns due to leakage across the edges of strip separators.

The leads from the winding are brought out of the gas-tight housing by means of semi-circular conductors running through the central inspection hole of the forging to the slip-rings.

Excitation is provided by an exciter running at 1,000 r.p.m. and driven through gearing from the main alternator shaft.

The bearings, which are of the usual spherically seated type, are carried in cast steel brackets bolted to the outer frame.

Shaft Seals

Leakage of hydrogen along the shaft is prevented by oil-sealed glands of the floating ring type. Two bronze rings are mounted side by side in an annular groove in a cast steel support which is carried rigidly from the inner surface of the bearing bracket. They are permitted to move with the shaft in a plane perpendicular to the axis of the rotor, but are prevented from rotating with it. The clearance between the rings and the shaft is kept down to the minimum consistent with satisfactory running. Similarly, the clearance between the rings and the side of the groove in which they are contained is kept small to reduce the oil flow as much as possible and to maintain a satisfactory seal at this point. The groove behind the rings is supplied with oil at a pressure of about 10 p.s.i. above that of the gas in the frame, and feed holes in the rings allow the oil to pass into a small annulus formed between the rings by machining the rings themselves. The film of oil thus maintained between the rings and the shaft forms the gas seal and at the same time lubricates and cools the rings.

The seals are supplied with oil which is normally vacuum treated to remove entrained air and water in order to prevent adulteration of the hydrogen from the oil which flows to the gas side of the seals.

The seal-oil treating unit is located at basement floor level within the foundation block and contains the various items of auxiliary equipment associated with the oil supply to the seals.

The Napier Spinning Rig

By P. J. WALLACE, Chief Designer, Test Plant Department, D. Napier & Son Limited.

The design and development of high-performance gas-turbine engines for aircraft has necessitated a great deal of research into the physical properties of materials; it has also been found desirable to proof-test all highly stressed components before final assembly. It was with a view to facilitating some of this work that the spinning rig was proposed. Among various research requirements was the need to investigate the behaviour of rotor discs when centrifugally stressed beyond the elastic limit of the material. For purposes of proof-testing it was required to subject every rotor disc to a period of over-speeding prior to assembly in the turbine.

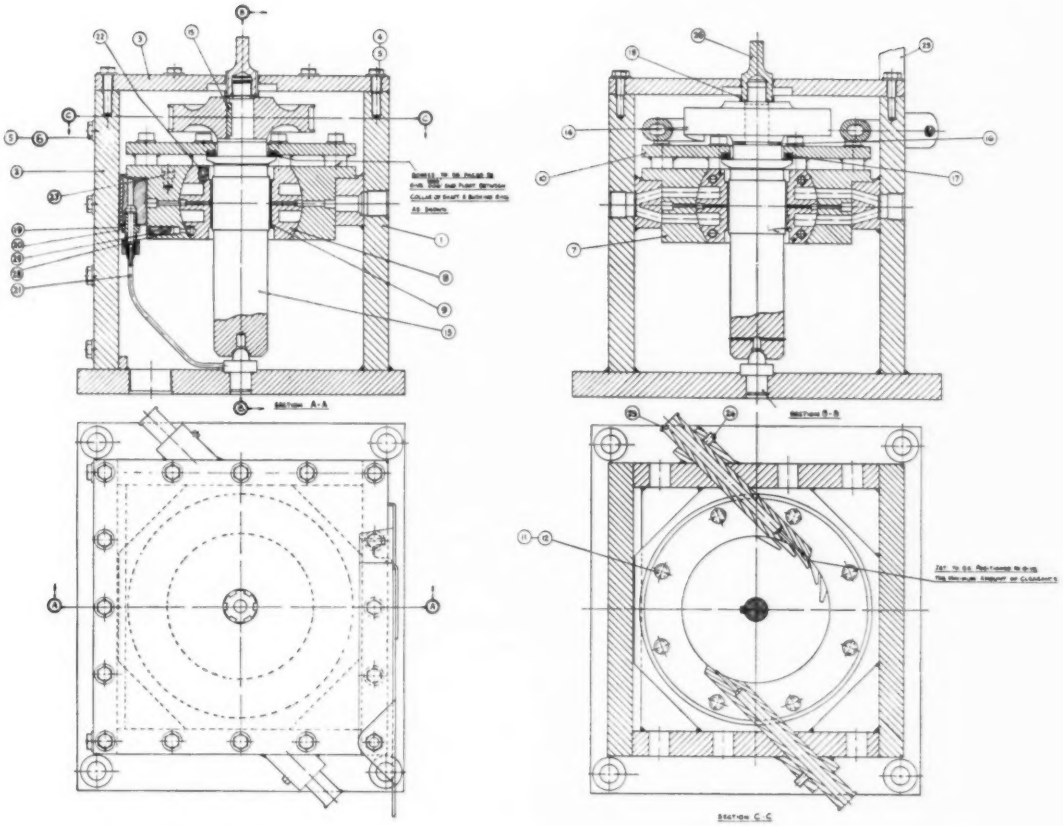
It became evident quite early in discussion that the whole programme could not be covered by a single rig, owing to the wide range of diameters of disc to be tested and the corresponding range of rotational speeds—extending from 7,000 to 75,000 r.p.m. Accordingly, a decision was taken to proceed, in the first instance, with a rig capable of handling discs up to 17 inches diameter and weighing up to 30 pounds, at rotational speeds varying from 7,000 to 30,000 r.p.m. It was realized that even this limited project would entail some very difficult problems and that in consequence it would be wise to postpone design of more ambitious machines until considerable running experience had been gained from the first.

Enquiries were made to ascertain what information, if any, was available regarding other spinning rigs which might have been built and run either at home or abroad. Very little was discovered; only two or three rigs were found to have been built, and information about their design or performance was extremely scant. It became clear that the only possible approach to the problem must be from first principles. At the very outset a fundamental decision had to be taken; either the disc under test would be mounted on a thin vertical shaft, sufficiently flexible to allow the relatively heavy disc to rotate about its own mass-centre, or on a very stiff shaft which could then be mounted horizontally or vertically according to choice. In the former case, the first, second and probably other critical speeds would lie within the running range, thus necessitating the provision of dampers to avoid the build-up of dangerous vibrations; in the latter case the shaft would be so designed as to place the first critical speed outside the running range. Which would prove the better choice was not easy to decide. The small-diameter shaft would simplify the bearing problem but, on the other hand, the development of adequate dampers might prove a protracted process of experiment.

It is a primary requirement of all special test-plant that it shall be capable of doing its job from the moment of its completion; if some experimental or development work is unavoidable then it is most desirable to endeavour to carry this out concurrently with the overall design and manufacture. This was felt to be impossible where dampers were concerned because the findings of the development work might necessitate extensive scrapping and revision of the rig design. On the other hand, if development work were to be initiated in connection with bearings for the stiff—and much larger—shaft, the design of the whole rig could go ahead in the certainty that subsequent modifications to accommodate suitable bearings could not be very extensive.

Experimental Bearing Rig

Preliminary calculations indicated that the bearings could not be less than about $1\frac{3}{4}$ inches diameter without impairing the stiffness of the shaft. It did not take long to establish that no available ball or roller bearings were likely to prove satisfactory. Enquiries of a well-known firm specializing in the manufacture of plain, or sleeve,



Ref. No.	*Description*
1	Casing
2	Front end
3	Top plate
4, 5 & 6	Bolt and washer
7	Housing split
8	Bearing sphere split
9	Vandervell bearing split
10	Retaining ring
11	Allen screw
12	Spring washer
13	Disc shaft
14	Disc turbine
15	Key
16	Shim

Ref. No.	*Description*
17	Bushing ring
18	Shrouded washer
19	Washer
20	Union body
21	Oil pipe
22	Ferrule
23	Air jet
24	Screw
25	Fulcrum pin
26	Adaptor
27	Shoulder dowel
28	Screwed locating pin
29	Socketed grub screw

Fig. 1. *Special bearing rig made for experimental and development work on the spinning rig*

bearings revealed the fact that the very high peripheral speed represented a problem so far unexplored. Consequently, it was decided that the Napier company should build a special bearing rig to permit the necessary experimental and development work. This rig is illustrated in Fig. 1. A shaft of suitable proportions was arranged to run (in a vertical position) in the sleeve bearing, a combined steady and thrust bearing being provided at the other end. Mounted above the test bearing was an impeller driven by air jets supplied from the workshop airline at a pressure of 60 lb/sq. in.

It was felt that no matter how stiff the shaft for the spinning rig might be made it would still flex to a degree, and that the bearings would need to be mounted in such a way that they could accommodate themselves to such deflection. The bearings were therefore carried in spherical shells mounted in split housings. Obviously such an arrangement demanded workmanship of a very high order. It will be apparent from the illustrations of the bearing that arrangements were made to circulate oil through grooves in the outer spherical surface of the shell in order to facilitate the transfer of heat from the bearing. This secondary oil circuit was of equal value in heating-up the bearing, shell and housing to somewhere near operating temperatures before starting-up, by circulating hot oil through the assembly. Unless the bearing, shell and housing be heated in this way there will be a risk of the heat initially generated at the bearing-surface causing the shaft to expand faster than the bearing, with consequent seizure.

Space does not permit a description of the bearing development which was carried out on this special rig. However, work was commenced with a " no-load " bearing and the conditions then made steadily more arduous until it was eventually found possible to run the rig for periods exceeding two hours at speeds of 45,000 r.p.m. and with a specific bearing pressure of 225 lb/sq. in.

Motive Power

Another basic decision concerned the motive power to be provided and the form which this should take. The need for economical running, apart from any other considerations, suggested the advantage of enclosing the test disc in a vacuum chamber. Clearly, if the chamber could be exhausted completely of air the requisite power would be merely that required to overcome mechanical friction. Estimates were made of the power likely to be absorbed in rotating the largest discs in various degrees of vacuum. Serious consideration was given to the possibilities of using an air turbine to drive the rig, particularly because this would permit a very simple transmission. Against this would have been the drawback of depending upon a shop airline, always liable to be overloaded and consequently subject to pressure fluctuations. Largely with a view to making the rig independent of an uncertain domestic service, it was decided to use a variable-speed electric motor. Exigencies of supply influenced the selection of a 100 h.p. machine.

This decision inevitably compelled the provision of a speed-increasing gearbox having an overall gear ratio of 20 to 1. This high ratio in conjunction with the characteristic curve of the motor gave rise to doubts as to whether there would be available at low speeds sufficient torque to cover all conditions of the research programme. It was therefore decided to add a pony motor of 25 h.p. and capable of high torque at low speed.

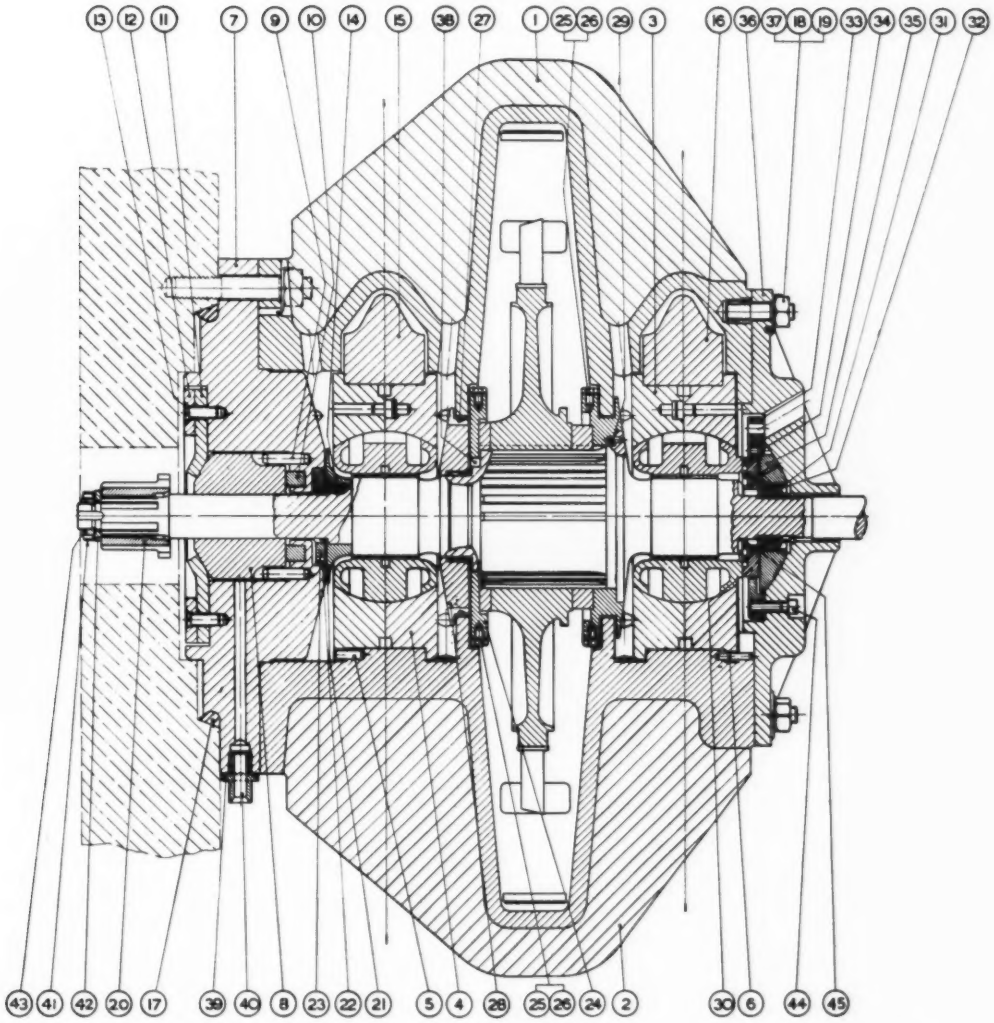
Seals for Bearings

Having decided to introduce a vacuum chamber, it became evident that the bearing problem would be accentuated in so far as seals would have to be added to prevent the escape of air along the shaft into the chamber. Steps were accordingly taken to elaborate the special bearing rig so as to permit suitable seals being developed.

The general form of the first type of seal will be apparent from the assembly of the spinning casing (Fig. 2). It comprised a special carbon block, highly finished in the bore to a close running fit on the shaft. In order to relieve the seal of any appreciable radial load due to malalignment, it was retained in position by a soft rubber diaphragm. This type of seal did not prove very reliable in practice and was later replaced by one of the plate type. This, in turn, is now being superseded by one of more novel design.

Other Problems

Before proceeding with a description of the spinning rig and its auxiliaries, mention should be



Ref. No.	*Description*	*Ref. No.*	*Description*	*Ref. No.*	*Description*
1	Casing, top half	15	Bearing cap, inner	32	Ring nut
2	Casing, bottom half	16	Bearing cap, outer	33	Pad retaining ring
3	Spherical bearing and housing, outer	17	Distance piece	34	Locating peg
4	Spherical bearing and housing, inner	18 & 19	Nut and tab-washer	35	Thrust pad
5 & 6	Locating pin	20	Sleeve	36	End cover
7	Carbon seal housing	21	Upper oil thrower	37	Stud
8	Carbon seal	22	Shrouded washer	38	Key
9	Locating peg	23	Ring nut	39	Washer
10	Inner carbon seal	24	Balancing ring	40	Union body
11	Seal	25 & 26	Screw and tab-washer	41	Shrouded washer
12	Seal retaining ring	27	Shrouded washer	42	Washer
13	Screw	28	Centre oil thrower	43	Ring nut
14	Key	29	Locating peg	44	Socket head screw
		30	Thrust ring	45	Key
		31	Tab-washer		

Fig. 2. Sectional view of spinning casing

made of various difficulties in the matter of lubrication which were foreseen in the early discussion of the project and which became the subject of special study while the design of the mechanical features went ahead. It was felt that difficulties would arise from the necessity of excluding all air from the oil circuit and, of course, from the oil itself. This would involve a pressure-tight pipe system and the submersion of the pumps within the oil tank; moreover, effective glands would be required around the driving spindles where these passed through the tank wall. Arrangements for controlling oil temperature would also be essential.

There was the further problem of obtaining accurate measurement of rotational speed of the disc.

It was also realized that a heavy piece of metal rotating at these very high rates could, under some accidental circumstances, become a lethal weapon dangerous alike to rig and personnel. In the first place, it appeared that seizure of a bearing might do extensive damage to the rig owing to sudden application of an indeterminately high torque upon the casing and supports. For protection it was therefore decided to provide two mechanically weak points—or "fuses"—one at the attachment of the spinning casing to its supporting frame, and the other in the transmission between gearbox and disc shaft.

A more serious situation would arise if the disc under test should fracture, thus giving rise to violent out-of-balance forces on the whole machine and, worst of all, to penetration of the casing by pieces of hot jagged steel travelling at velocities more usually associated with projectiles. To protect operating personnel from this danger it was decided to provide an adequate shield of armour-plate.

This brief outline of the inception of the project is now followed by a description of the construction of the rig.

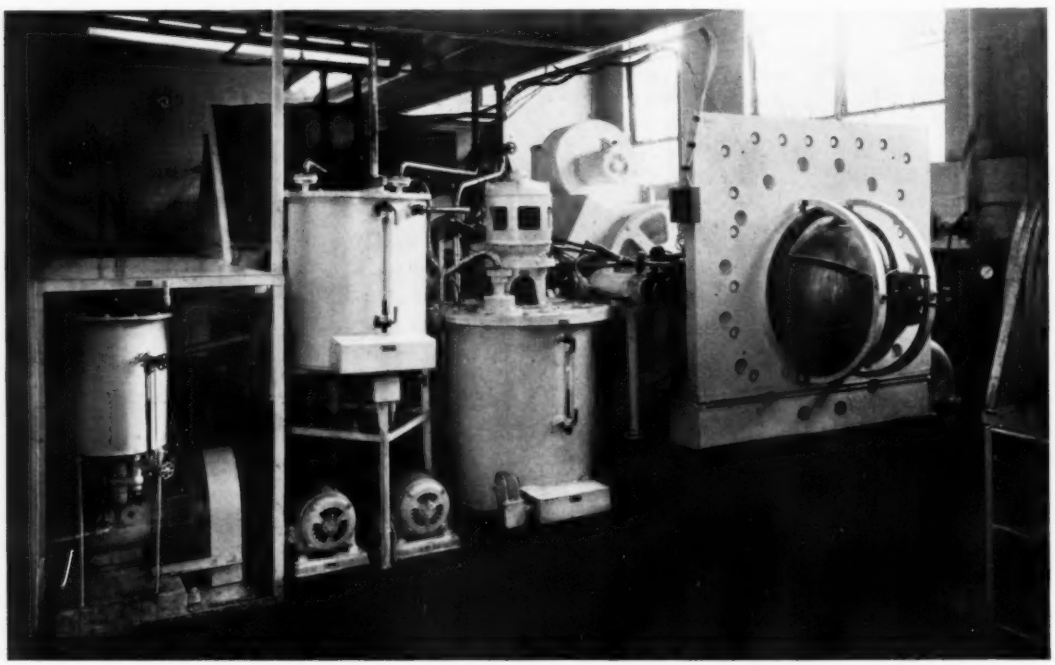


Fig. 3. General view of spinning rig, with domed cover enclosing the rotor clamped to armour-plate bulkhead on right

The Rotor Casing

The general shape of the rotor casing will be clear from Fig. 2. It is split in the axial plane so as to permit inspection of the entire rotating assembly in position. For this purpose the caps securing the outer spherical bearing-housings are made independent of the top half of the casing which, in fact, becomes nothing more than a cover to the lower half. A well-ribbed aluminium casting was chosen, slotted at intervals around its maximum periphery to permit ready escape of any oil which might enter the inner chamber. The complete casing is mounted as a cantilever upon a stout vertical bulkhead of armour-plate, a double-ended spigot-plate being interposed to ensure correct alignment. The ring of fixing bolts have been so proportioned as to hold the casing rigidly during normal running, yet to be easily sheared in the event of violent bearing-seizure; by this provision it is hoped in such event to limit damage to the relatively cheap casing, and to preserve the expensive assembly of shaft, bearings and housings.

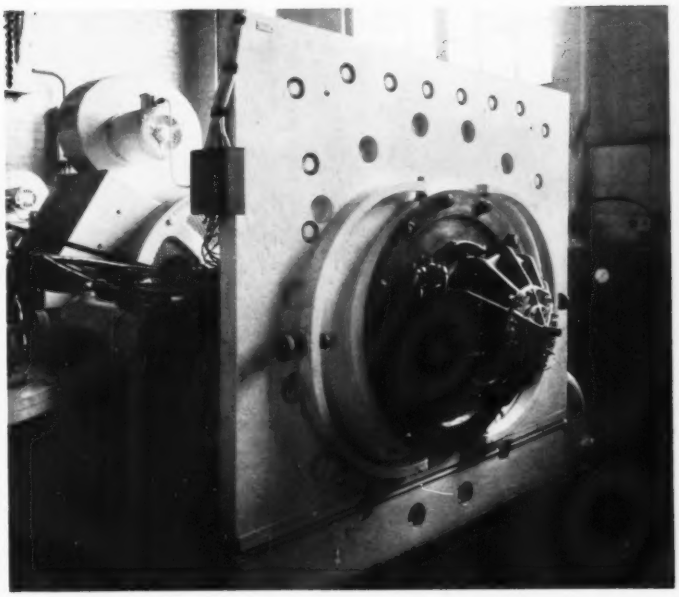


Fig. 4. Rotor casing with cover removed and hinged clamping rings swung back to the bulkhead

The Rotor Shaft Assembly

A number of shafts have been made, of various dimensions at the splined portion, so as to accommodate the wide variety of discs required to be handled. In the interest of economy, distance collars are provided to suit different hub-widths of the discs. Adjacent to each of these collars is a balancing ring carrying a number of radially mounted set-screws capable of ready adjustment during the process of dynamic balancing the rotating assembly. These set-screws are locked by means of tab-washers. Next to the balancing rings (in one case made integral with the ring) is an oil-thrower with the effective diameter so disposed as to fling off any excess of oil through a series of holes in the casing wall. The whole combination of disc, collars and rings is sandwiched between a shoulder and a slotted ring-nut which is made to serve as one of the oil-throwers. Each member is keyed or dowelled to prevent relative creeping. The ring-nut is locked by means of a shrouded washer.

Beyond the bearing on the gearbox end of the shaft (left-hand in Fig. 2) is another oil-thrower, shrouded-washer and nut. The illustration shows the original type of carbon seal. This comprised two parts; the larger member to prevent leakage of air along the shaft surface, and the small ring to prevent leakage past the outer surface of the larger seal. The material used for these seals was known to be very fragile and it was considered essential to avoid any unnecessary stress. For this reason the larger seal was given ample clearance on its outer diameter so as to allow it to move freely in any way which the shaft might dictate. A pair of loose-fitting dowels served to prevent rotation, and the seal was restrained axially by a rubber diaphragm. As already stated, this type of seal did not prove very reliable owing to spasmodic breakage.

Axial location of the rotor assembly is effected by a double-thrust washer at the right-hand end of the shaft. Owing to the spherical mounting of

the journal bearings, the static thrust-ring is necessarily left free to float on its own spherical seating. The shaft is extended beyond the thrust bearing to carry a number of collector rings for use in strain-gauging. Provision is made in the shaft for housing the leads to the test-disc, while the associated brush-gear is carried on an extension to the casing.

Vacuum Casing

Under operating conditions, the entire rotor casing is enclosed within a domed cover tightly clamped to the armour-plate bulkhead, as seen in Fig. 3. A rubber sealing-ring maintains an air-tight joint and the half-hoop clamping rings are hinged to the bulkhead to give maximum access to the rotor casing when the dome is removed, as in Fig. 4.

Transmission and Driving Motors

Various elements of the entire transmission are shown diagrammatically in Fig. 5.

The speed-increasing gearbox is of straight-forward design with twin layshafts to balance out the loads on the high-speed pinion bearings. All the gears are spur type; ball or roller bearings are used throughout with the exception of those for the high-speed pinion which are practically identical with those used on the rotor shaft and are spherically mounted.

The connecting member between the terminal gear-shaft and the rotor shaft has been the subject of some experiment. The problem here has been to provide a link which can be relied upon to transmit maximum torque without interruption and yet readily fracture in the event of any substantial increase. In addition, it is required that this connecting member shall accommodate very slight malalignment between gear-box and rotor shaft. Although the gear-box was line-bored with the spigot-bore in the bulkhead, some mal-alignment under dynamic conditions is inevitable.

The original design, partly shown in Fig. 2, comprised an internally splined tube of small diameter engaging with sleeves splined both inside and out and fixed to their respective shafts. The fit of the outer spline was made such as to provide axial freedom and minute angular displacement. The purpose of the intermediate, double-splined sleeves was to avoid the necessity of renewing an entire rotor shaft due to fretting of loose-fitting splines.

The normal routine use of the rig necessitates frequent mounting and removal of the entire rotor casing assembly, and with so fragile a transmission it would be very difficult in ordinary handling to avoid either burring or bending the shafts. Provision has therefore been made for small platforms to take the weight of the casing when first offered up; in addition, long dowel-pins are attached to the bulkhead for the purpose of bringing the casing into correct alignment before the splines can come into engagement.

Behind the gearbox, at the low-speed end, is the 100 h.p. variable-speed A.C. motor having a speed range of 375 to 1,400 r.p.m. Between the motor and gearbox is a flexible coupling which incorporates rubber bushes. As already mentioned, the torque characteristics of this motor at low speed are such that it was found necessary to add a pony motor of 25 h.p. for starting the rig. This motor has a speed range of 500 to 4,200 r.p.m.; consequently a speed-reducing gearbox, with a ratio of 9.9 to 1, is fitted between the two motors. This gearbox is connected by a dog clutch to the 100 h.p. motor.

The procedure in starting the rig is first to engage the clutch and then to start the 25 h.p. motor; this is allowed to accelerate up to 3,800 r.p.m. when the larger motor is switched in and takes

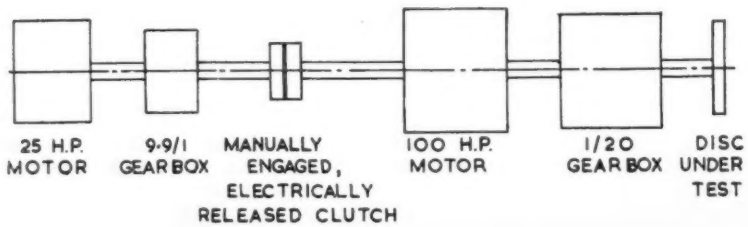
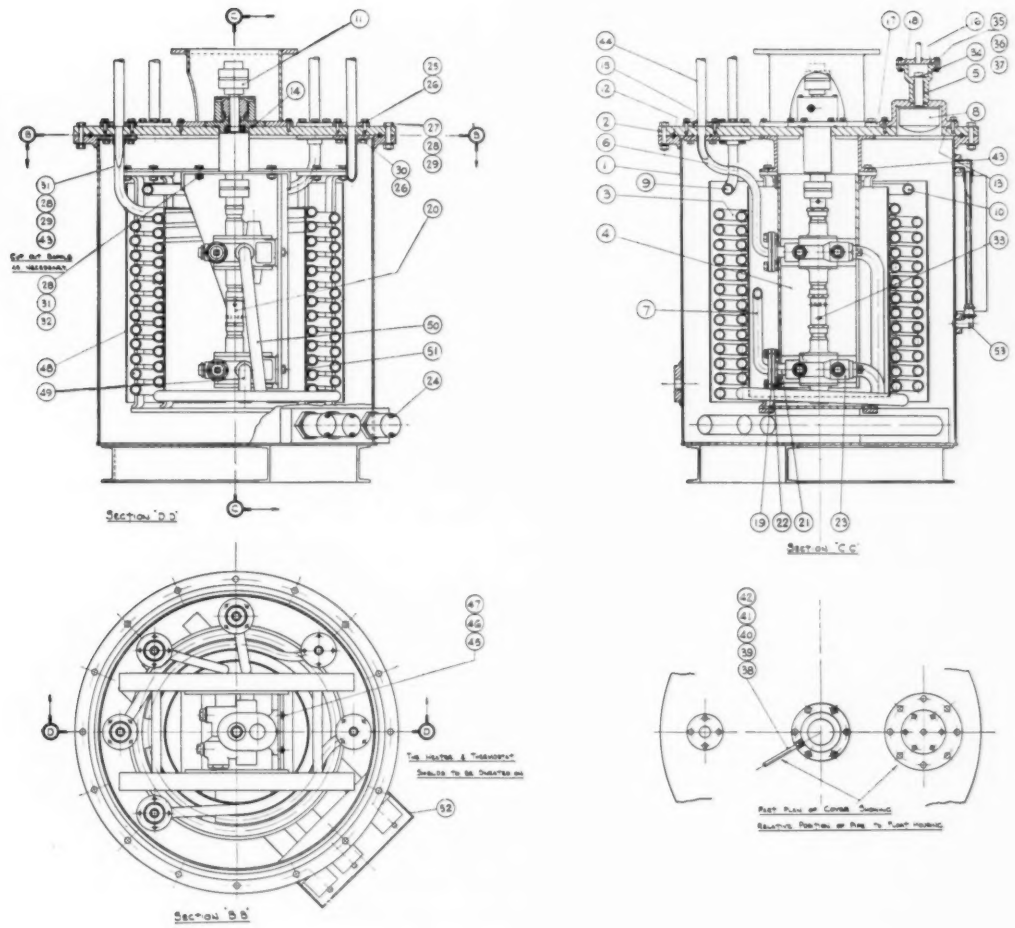


Fig. 5. Diagrammatic arrangement of drives and transmission to disc under test



Ref. No.	*Description*
1	Tank
2	Cover
3	Cooling pipes
4	Pump bracket
5	Float housing
6	Outlet pipe, top
7	Outlet pipe, bottom
8	Float
9	Oil inlet pipe
10	Oil inlet pipe (long)
11	Pump drive
12 to 16	Sealing rings
17	Blanking flange
18	Float housing flange
19	Joint washer
20	Coupling sleeve

Ref. No.	*Description*
21	Adaptor
22	Flange
23	" Albany " pump
24	Heaters
25	Bolt
26	Spring washer
27	Bolt
28	Nut
29	Spring washer
30 & 31	Bolt
32	Washer
33	Pin
34	Circlip
35	Bolt
36	Nut
37	Spring washer

Ref. No.	*Description*
38	Union body
39	Washer joint
40	Nipple
41	Nut union
42	Pipe
43	Tapered washer
44	Pipe
45, 46, 47	Bolt, nut and washer for " Albany " pump
48	Baffle
49 & 50	Suction pipes
51	Inner baffle
52	Heater cover
53	Elbow, tube, joint sleeve and jubilee clips

Fig. 6. Vacuum oil tank containing two pumps for lubricating the spinning casing

over the drive. The clutch is disengaged by means of a solenoid-operated mechanism, and it is the operator's responsibility to ensure that the pony motor is not allowed to exceed a speed of approximately 4,800 r.p.m.

Oil System

There are in effect three separate oil systems.

Taking the smallest one first, the small tank shown at the extreme left of Fig. 3 is used solely for the purpose of providing oil seals for the vacuum pumps.

The large oil tank on the right is used for the lubrication of the spinning casing. As already mentioned, it is essential that all air be exhausted from the oil, and the entire oil system kept under a vacuum. In order to facilitate these conditions it was decided to place the oil pumps within the actual tank. Fig. 6 shows a cross-section of the vacuum oil tank. It will be seen that two pumps are mounted end-to-end and driven by a vertical-spindle motor mounted on top of the tank; thus only one gland is required for the driving shafts. One pump is used for delivery of oil to the high-speed bearings of the spinning casing, and the other for supplying the cooling oil which circulates round the outside of the spherical bearing housings. All pipe connections are made in the detachable top of the tank. Oil temperatures within the tank are controlled by means of electric heaters and a coil of copper pipe through which cold water may be circulated.

It will be evident that, owing to the very high degree of vacuum existing within the casing, the scavenge pump can operate only by virtue of the force of gravity. Consequently the draining passages have been made of very large cross-sectional area. The scavenge pipe, which can be seen in the lower right-hand corner of Fig. 3, connects to a scavenge tank placed at floor level. Here again the scavenge pump is placed within the sump tank so that the whole system may be enclosed within the vacuum, thus avoiding the use of another seal or gland. Two filters are provided in this oil system; one between the sump and the oil tank, and the other between the oil tank and the rig-casing.

The middle of the three oil tanks is used for the lubrication of the gearbox. As already explained, the high-speed pinion of the gearbox is carried in two spherically-mounted plain bearings practically identical with those used in the spinning casing. The oil supply to these assemblies is also similar; oil is fed separately both to the bearing surfaces and to the cooling channels around the spherical housings. A branch in the delivery line feeds oil to jets within the gearbox, which spray it over the main gear teeth. These jets are arranged close to the point of meshing and on the leading side. The remaining bearings in the gearbox are either ball or roller type, and these rely for their lubrication upon oil splash.

It need hardly be mentioned that extreme care was taken with both the design and the fitting of all oil system components in order to reduce the possibility of air leakage to an absolute minimum.

Vacuum Pumps

Two vacuum pumps are used; one for exhausting the chamber containing the spinning casing, and the other exclusively for the oil system. These are of a standard proprietary design and have proved quite satisfactory in service. A problem which arose in the course of design was to assess the probable leakage of air, and so to determine the capacity of vacuum pump required. In the event, the matter was largely settled by the limited range of pump capacities available. It was fully realised that the choice of pump capacity depended solely upon the rate of air leakage, the degree of vacuum within the casing depending solely upon the efficiency of the vacuum pump.

Safety Precautions

These fall under two distinct headings; the protection of operating personnel from serious or even fatal injury, and the preservation of an expensive piece of equipment. The discussion of safety precautions is always a difficult matter; so much depends upon the temperament and the imagination of the parties concerned, quite apart from the question of knowledge. In the present case the difficulties were accentuated by the fact that very little information indeed was available of what might happen in the event of fracture or disintegration of the rotating parts. However, all parties to these discussions were agreed on the need of reducing risks to personnel to an absolute

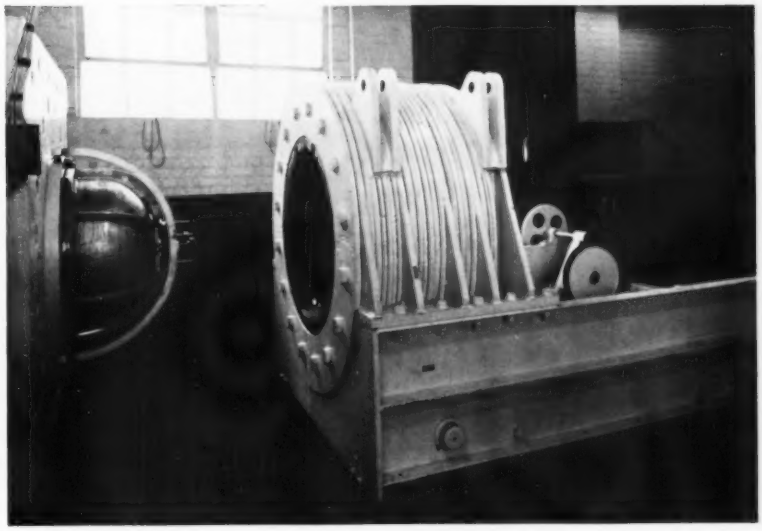


Fig. 7. Protection tunnel withdrawn from the rig on its carriage

minimum. This attitude has been fully justified by reports which have since come to hand of events, occurring elsewhere, involving the fracture of parts rotating at a very high speed. It is gratifying to report that up to the time of writing there has been no occasion to test the safety precautions provided on the Napier Spinning Rig.

The only relevant fact known at the time when the design was under consideration was that a large impeller of a centrifugal compressor had, on bursting, succeeded in denting a protection tunnel comprising laminations of high tensile steel having a total radial thickness of 11 inches. The use of various materials for a tunnel was discussed including concrete, steel, paper, wood and finally, armour-plate. The exigencies of space compelled the selection of armour-plate and it was first proposed to use this in a laminated form up to the same total thickness of 11 inches. Supplies of armour-plate were found to be available from a tank disposal depôt but before deliveries could be completed restrictions were placed upon the supply of this material with the result that the quantity originally intended could not be obtained. Moreover, there was considerable difficulty in obtaining facilities for rolling armour-plate varying from $1\frac{1}{2}$ to $2\frac{1}{2}$ inches thick. This compelled changing the design from one where the laminations were arranged co-axially and rolled to fit one over the other, to one where the plates were left flat, cut to a common shape and then arranged side by side along the length of the axis. The limited supply of armour-plate restricted its use to what was considered to be the most vulnerable portion of the length of the tunnel, the remaining plates being made from mild steel of such thickness as was readily obtainable.

Interposed between the steel plates are sheets of lead, the whole assembly being joined together by a number of 2-inch diameter bolts. In the event of any fracture of the disc under test it would obviously be most desirable to recover in the best condition the individual pieces for their evidential value. For this reason the steel protection tunnel is lined with several inches of hard wood arranged to leave only a small clearance beyond the vacuum cover. For freedom of access to the spinning casing it was necessary that the protection tunnel should be capable of being moved away from the rig when not in use. The total weight of the tunnel being about 10 or 11 tons, it was mounted on large-diameter flanged wheels which run on steel rails flush with the concrete floor. Movement of the tunnel and its carriage is effected by means of a simple train of gearing giving a high reduction ratio; one man can quite easily move the entire apparatus (Fig. 7).

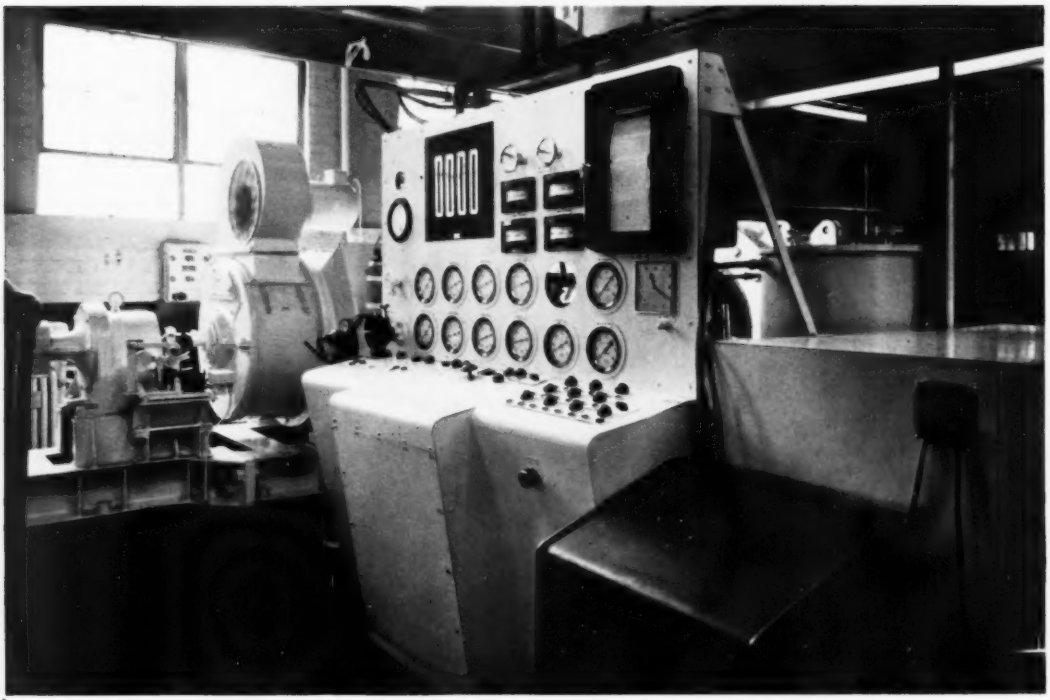


Fig. 8. The control desk on which all instruments and controls are centralised

Under running conditions, the spinning chamber is filled with oil-mist and it was considered that there would be a considerable risk of explosion in the event of sudden penetration of the vacuum casing. Consequently, provision has been made for the automatic injection of nitrogen into the casing in immediate response to any sudden rise of pressure. Various electrical interlocks are provided in order to protect the machinery from careless handling; most of these are of a conventional nature not calling for any special description. Situated in the centre of the control panel is an emergency button to give rapid shut-down of the rig; this can be arranged to bring about simultaneous injection of nitrogen.

Control Desk

As will be seen from Fig. 8, all instruments and controls are centralised on the control desk. Of primary importance in the instrumentation of any spinning rig is the accurate indication of rotational speed. Three means of measurement have been provided: a Smith's speed-indicator driven through a gear reduction on the gearbox; a point for using a Hasler indicator; and for accurate routine use, an electronic speed indicator. This last instrument is of proprietary manufacture but subsequently developed by the Napier Gas Turbine Division; it gives a direct reading from the high-speed shaft of the gearbox.

Results Achieved

In the relatively short period during which the rig has been in operation, very valuable results have already been obtained. These have permitted considerable improvement in the design of rotor discs and other components, resulting in the more economic use of material and reduction of stress. Considerations of security preclude the publication of more explicit details of results but it may be stated with confidence that the outlay involved has been justified.

Tekapo Hydro-Electric Plant

By H. L. BAZALGETTE, A.C.G.I., M.I.E.E., A.M.Inst.C.E.

In July 1951 the Tekapo power station in the South Island of New Zealand, for which the generating plant and equipment were supplied by The English Electric Company to the order of the New Zealand Government, was brought successfully into commission.

The Tekapo project forms part of the extensive hydro-electric development of the Waitaki River in South Canterbury which is estimated to have a potential of at least 1,000,000 h.p., or about the same as the Dominion's present total output.

The existing Waitaki station with a capacity of 75,000 kW was opened in 1934 and consists of five 15,000 kW units, of which the fifth was installed in 1949. Two of the turbines and all five of the generators in this station were supplied by the Company which had thus already taken a prominent part in this important development. Two more units are now being added making seven totalling 105,000 kW.

Fig. 1 shows the electrical transmission systems in the South Island of New Zealand and the principal generating stations, including Tekapo and Waitaki.

Nearly 30 miles up the river from Waitaki station is Black Jack's Point where the Benmore project, one of New Zealand's greatest hydro-electric schemes, has yet to be carried out.

Lakes Pukaki and Tekapo together form the source of the Waitaki River, and dams are being built to control these snow-fed lakes so that spring and summer water can be stored for release during winter and also to regulate the supply of water to the Waitaki station lower down.

At Tekapo, the power station houses a 25,200 kW vertical-shaft generating unit. The water feeding it is taken from the lake through a tunnel 19 ft. in diameter and 1 mile long to a 160 ft. diameter surge chamber. Thence the water passes via a single head-gate into a steel-lined penstock 20 ft. in diameter and 200 ft. long which is connected directly to the turbine without the intervention of any further gates or valves.

The station (Fig. 2) is located at an altitude of about 2,300 ft. where temperatures down to zero Fahrenheit are experienced, and the plant was designed with this factor in mind. As the single-unit arrangement has been adopted, the present equipment constitutes both the initial and the final installation. Starting is effected under local manual control, but full protective features have been provided as for an unattended station.

The main turbine at Tekapo operates under a gross head of from 80 to 105 feet, corresponding to net heads of from 71 to 96 feet, and develops 30,000 h.p. and 35,000 h.p. respectively under these heads, running at 150 r.p.m. It drives a 28,000 kVA 0.9 power factor 3-phase 11,000 volts 50 cycles "umbrella" type generator equipped with main and pilot exciters and an auxiliary alternator on the same shaft for electric governor drive.

In addition to the main generating plant, the contract included a 500 h.p. auxiliary generating set of the horizontal reaction type, the main and service transformers, 11,000 volts and 400 volts switchgear, control gear and protective equipment.

Fig. 3 shows the interior of the power station with the main generating unit, the governor actuator with pumping unit and gauge panel and also part of the auxiliary generating set with its equipment. On the floor below are located the guide vane servo-motors. The station is served by a crane of sufficient capacity to handle the heaviest lift during assembly, namely 175 tons.

Main Turbine

The main generating plant, of which a cross

Fig. 1. Electrical transmission systems and principal generating stations in South Island, New Zealand. 'English Electric' hydro-electric plant is installed at Waitaki, Tekapo, Highbank, Coleridge, Arnold, Cobb, and Monowai.

(Map copyright of Geographical Projects Ltd.)

Fig. 2. Tekapo hydro-electric power station

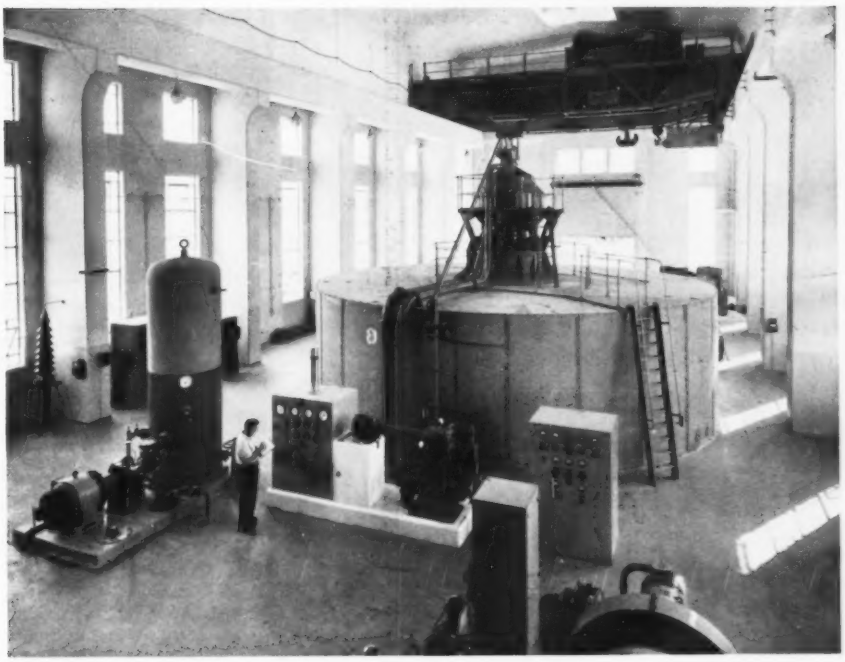


Fig. 3. Interior of Tekapo power station, showing main generating unit

section is shown in Fig. 5, is of particular interest since it includes a turbine of the propeller type having a runner with feathering blades and operating gear of the 'English Electric' patented design. This type of runner has special importance at Tekapo in order to give good efficiency over the wide range of operating head. The completed runner at Rugby Works is shown in Fig. 4, and a sectional model of the turbine and runner in Fig. 6.

In the design of the operating gear, complicated mechanism is eliminated with a resulting reduction in wear and backlash. The blade operating levers are simultaneously actuated by a powerful mobile servo-motor cylinder conveniently accommodated within the runner hub. This mobile cylinder surrounds a stationary piston which is rigidly bolted at the top of the runner hub. The runner blade shafts and their operating levers are rigidly connected and so arranged that the blade shaft bearings are liberally dimensioned, thus providing low specific pressures on these bearings in service.

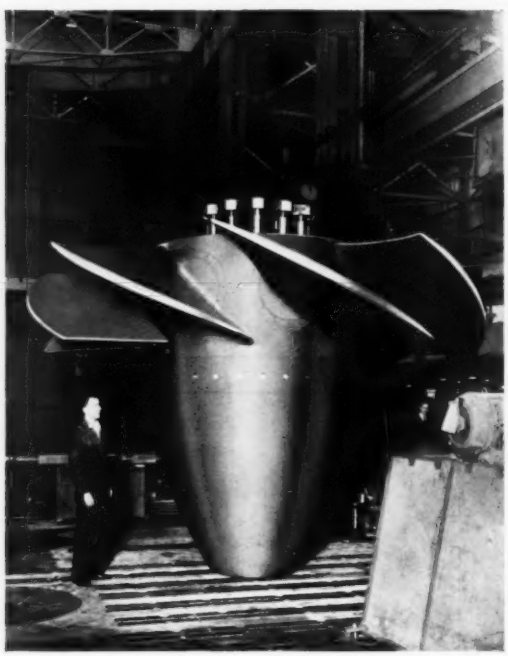


Fig. 4. The runner for the 35,000 h.p. turbine, completed at Rugby Works

This arrangement dispenses with the necessity for fitting a heavy and cumbersome servo-motor between the turbine and generator shafts. It has the further advantage that the servo-motor is placed in the immediate neighbourhood of and below the runner blading, so that the main guide bearing of the turbine can be located quite close to the runner wheel, thus obtaining great rigidity and freedom from vibration.

The mobile servo-motor is of original design. It is cylindrical inside and prismatic outside, the number of faces of the external prism corresponding to the number of runner blades. In each face of the prism is an inclined groove in which slides a bronze block connected by an adjustable pin to the blade operating lever. Axial guidance of the servo-motor cylinder is provided by massive radial lugs which fit into guide grooves machined inside the runner hub cover, thus taking care of the torque transmitted to the runner blade levers. The whole of the operating mechanism is immersed in oil.

One of the virtues of this operating gear is the ease of inspection and overhaul without the necessity for major dismantling. First the oil is drained off by the valve located at the bottom and in the centre of the nose piece. To expose the connections and operating parts it is then only necessary to lower the runner cover bodily, the runner itself still remaining suspended in the normal position. The operating gear and servo-motor are then readily accessible from below.

The hydraulic design of the runner was based on the results of very full model tests carried out at the Company's Hydraulic Laboratory at Rugby. The model runners were 20 inches in diameter and great care was taken to shape the blades very exactly. Moreover, the runner was tested in a spiral setting and with a draft tube bend closely following the shape of the full-scale installation. The output of the models reaches 45 h.p.

The blades of the 35,000 h.p. feathering propeller are of stainless steel and are accurately ground and polished all over, an important feature for securing high efficiency.

The turbine is provided with a spiral casing having an inlet diameter of 17 ft. 6 in., fabricated

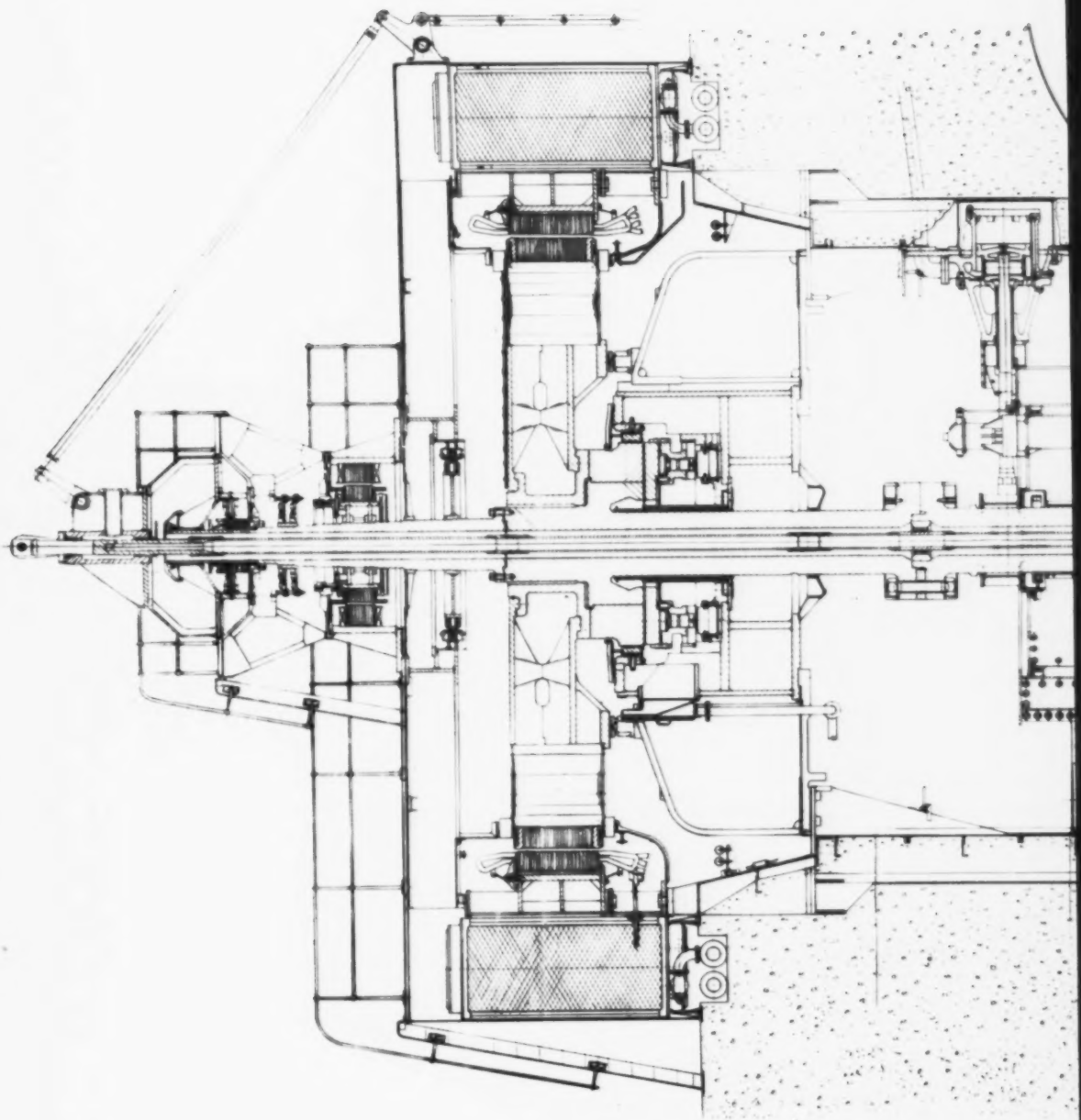
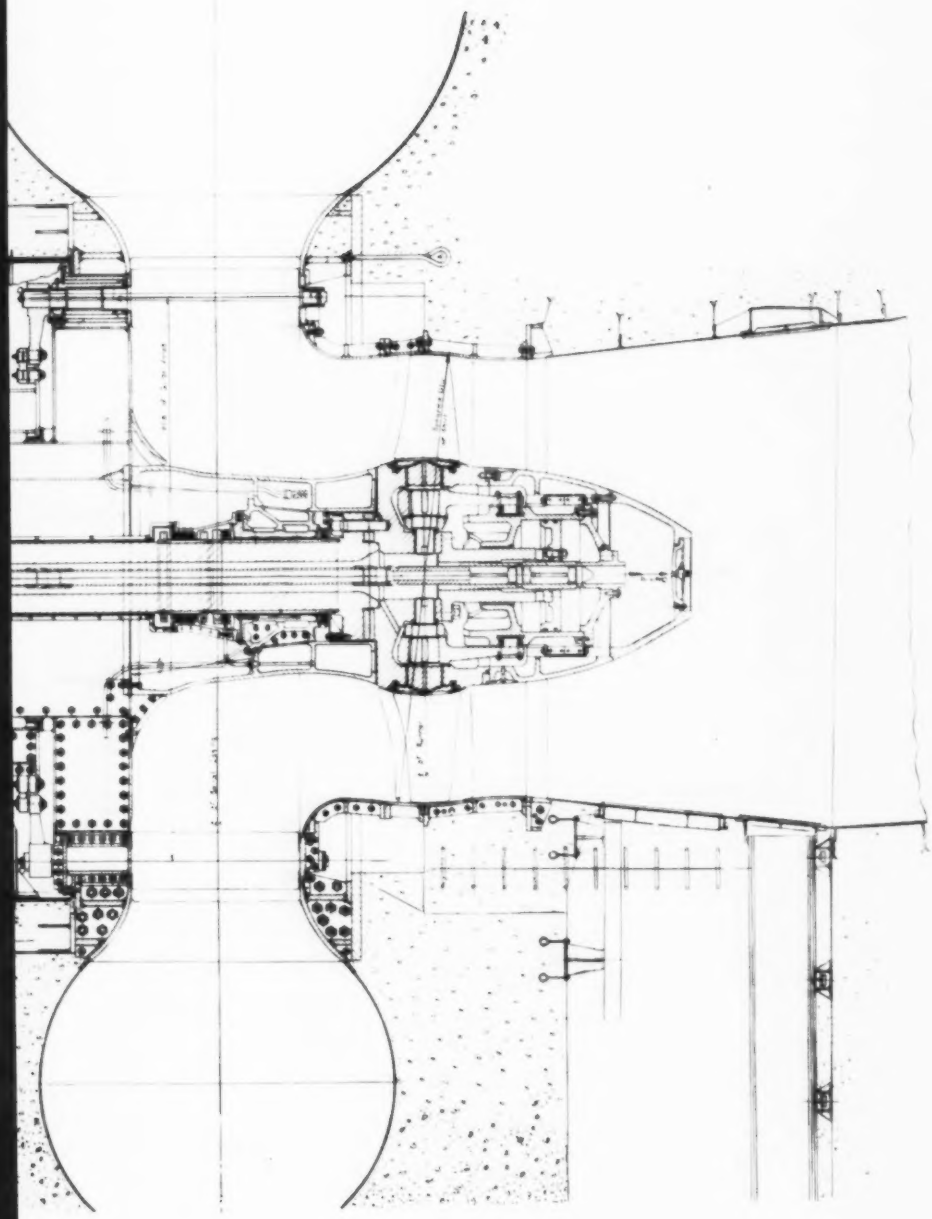


Fig. 5. Section through main turb



ne and generator

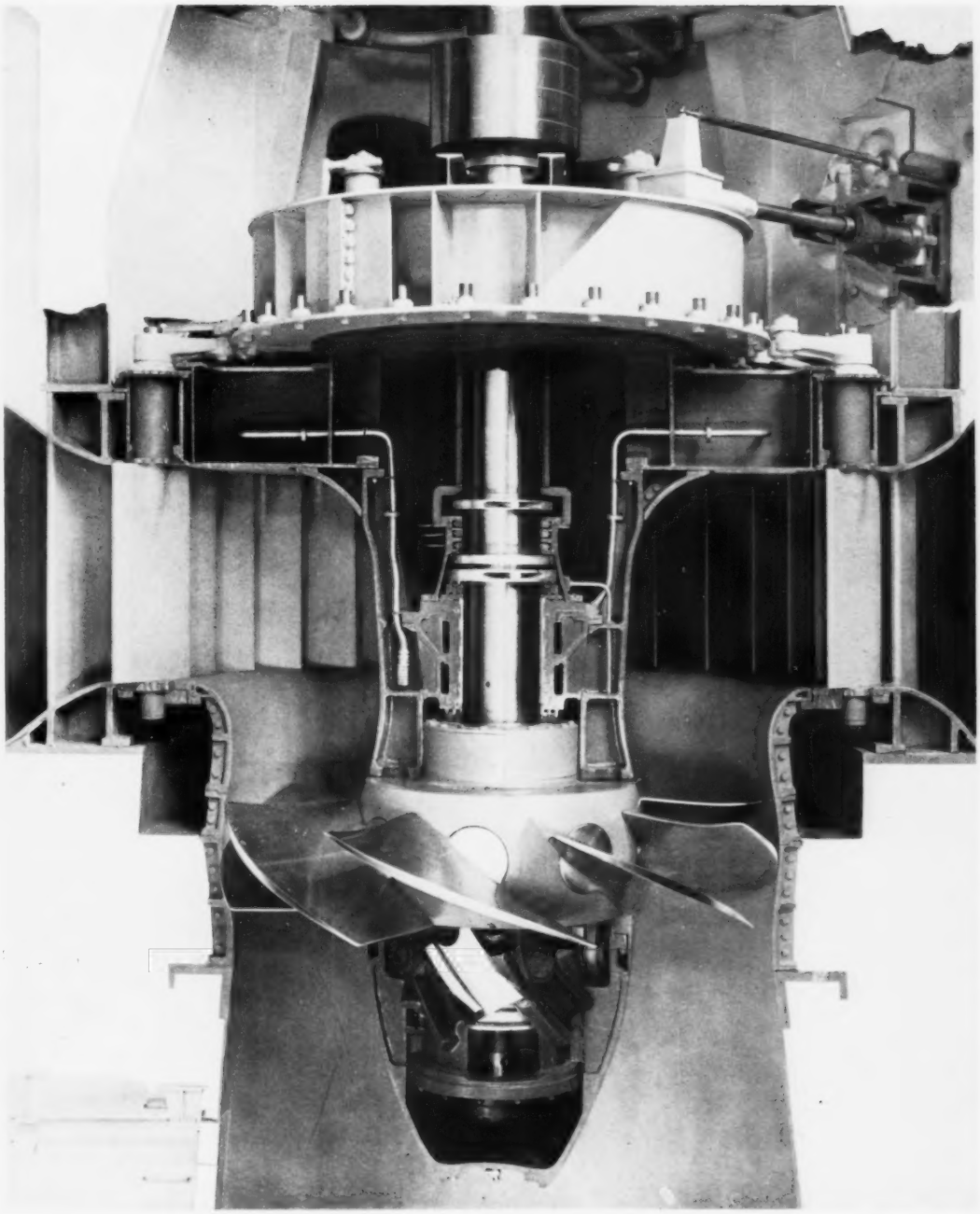


Fig. 6. Part of a sectional model, showing the main turbine and method of operating feathering blades on runner

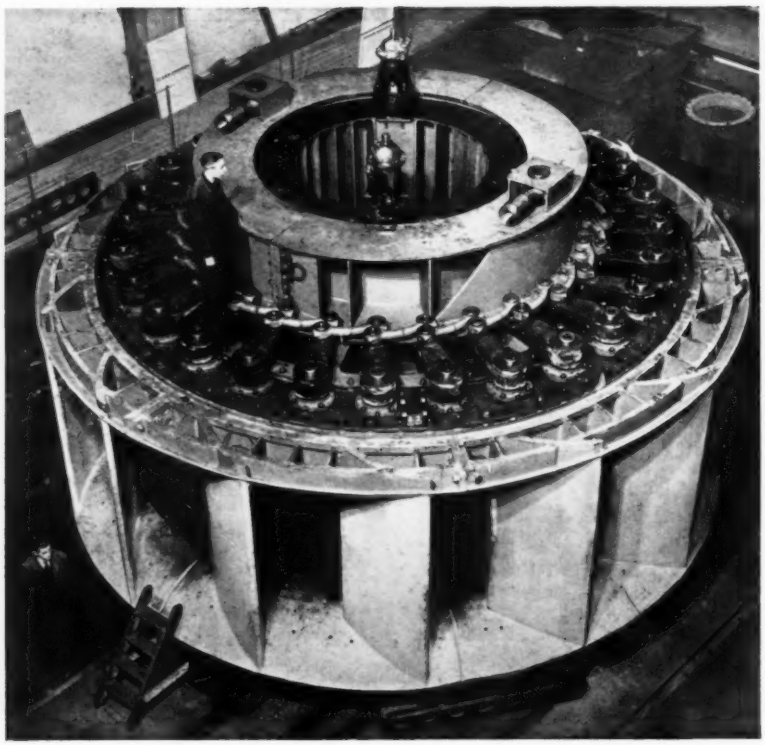


Fig. 7. Speed ring and guide vane assembly during erection at the works

by electric welding from steel plates varying in thickness from $\frac{7}{8}$ in. to $\frac{3}{8}$ in. according to the load sustained. In order to save transport, this major part was manufactured entirely in New Zealand and despatched in sections to the site where the final welding together took place. The speed ring (Fig. 7), which bridges the throat gap of the spiral casing, was made in England and assembled with the spiral casing at site and the whole welded together.

Other important fabricated parts in the turbine construction are the runner envelope in stainless steel, and the main cover, the regulating ring, the draft-tube liner and the generator support, all in mild steel. The runner envelope is spherically shaped, ensuring a small running clearance between the blades and the envelope for all blade positions, and gap cavitation is thus avoided.

The generator support is a massive structure which is bolted to the speed ring and transmits the generator load direct to the foundations, fulfilling the Company's established practice of ensuring the utmost rigidity and freedom from vibration in vertical-shaft units. The generator support together with the draft-tube liner were manufactured in New Zealand.

The turbine bearing is grease lubricated and consists of a cast iron housing of box construction containing a cast iron bush lined with anti-friction white-metal and cooled by water passing through a surrounding jacket. The lubricating system comprises two electric-motor driven pumping sets, each capable of supplying all the grease used by the bearing. By means of selector gear either pump may be chosen for normal or standby service, the standby unit starting automatically in the event of failure of the normal unit or excessive temperature in the turbine bearing.

The governing system is of the Company's usual oil pressure type with standard actuator and

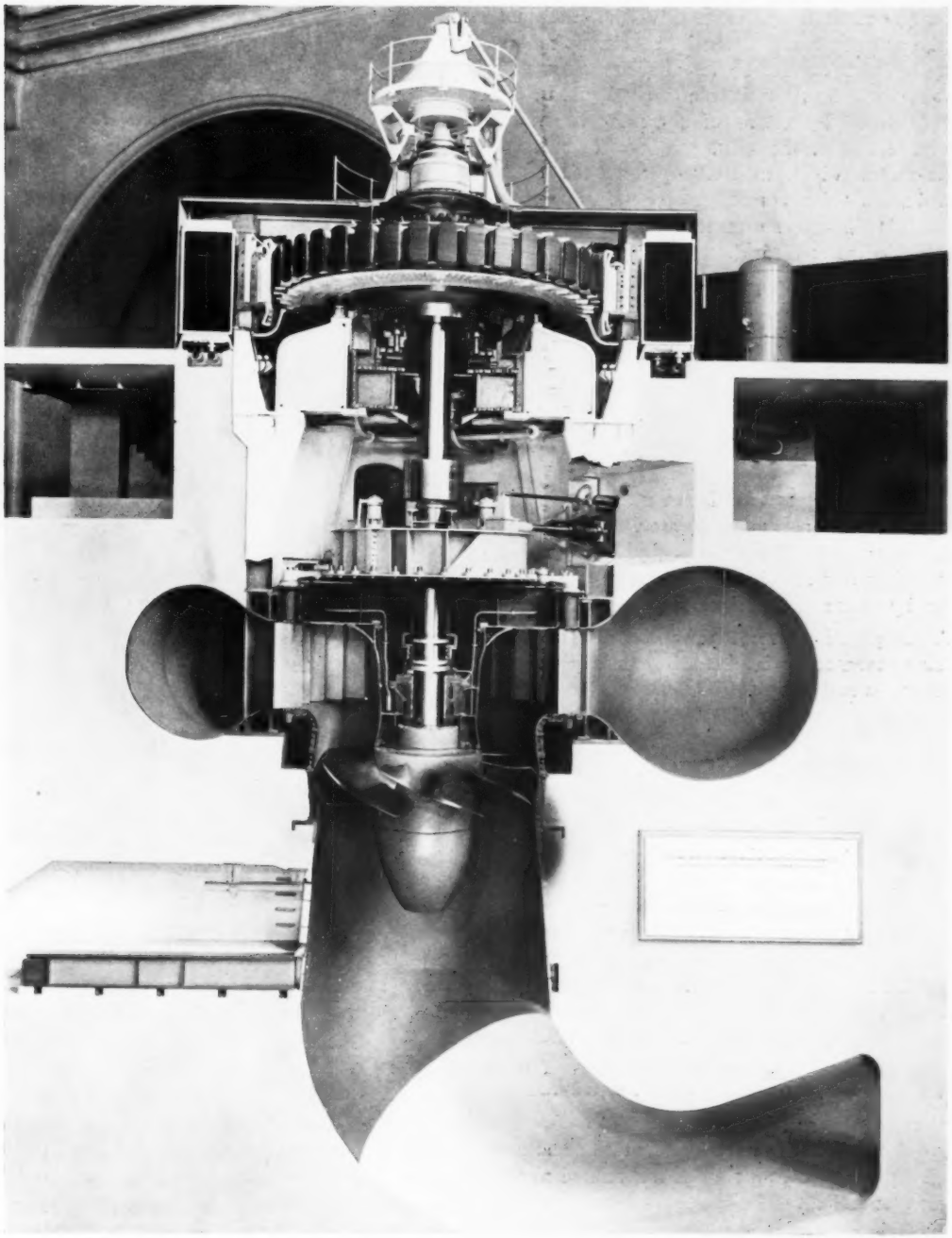


Fig. 8. Sectional model of the main generator and turbine

synchronous electric governor drive, the pendulum of the actuator being driven by an electric motor supplied with current by an auxiliary alternator mounted on the main alternator shaft just below the exciters. By means of a distributing valve in the actuator, the pilot governor operates a servo-motor which in turn operates the main distributing valve controlling the oil supply to powerful twin servo-motors for opening and closing the turbine gates through connecting rods.

The governor also performs its other function of controlling, by oil pressure, the runner blade servo-motor through a double-acting and compensated distributing valve located in the runner hub. Oil is fed to the connecting head situated on the top of the set, then down through piping in both generator and turbine shafts to the runner servo-motor.

Pressure oil is provided by an oil pumping set, electrically driven, and is conserved by an oil pressure receiver of large capacity, the function of which is to maintain the pressure at its correct value by means of a compressed air cushion.

A compact instrument panel located near the turbine governor carries the turbine and generator instruments and gauges.

Main Generator

The main generator is of the vertical-shaft " umbrella " type and has a maximum continuous rating of 28,000 kVA, 25,200 kW at 0.9 power factor, 11,000 volts 3-phase 50 cycles. It runs at 150 r.p.m. and is designed to withstand the turbine runaway speed of 368 r.p.m. Fig. 8 shows a sectional model of the generator and turbine.

The umbrella type generator has only one combined thrust bearing and guide bearing which is situated below the rotor and mounted on the power bracket so that the vertical distance between the centre-line of the guide bearing and the plane of the centre of gravity of the rotor is reduced to a minimum. Other advantages are :—

(*a*) The load due to the hydraulic thrust and the weight of the revolving parts is transmitted direct to the foundations.

(*b*) Only a light upper bracket is required to carry the exciter magnet frames.

(*c*) The light upper bracket and the absence of a bearing above the rotor results in a reduction in the height of the set compared with the two-bearing type.

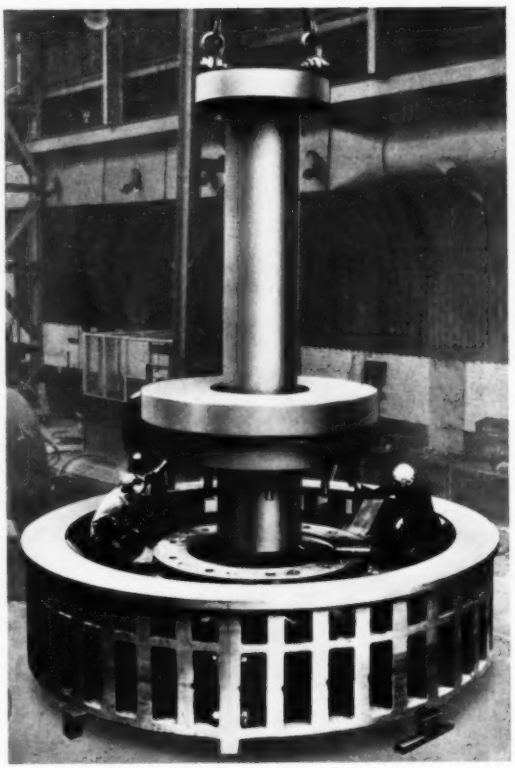


Fig. 9. The rotor hub, into which the generator shaft is being lowered

(*d*) Economy in head-room results from the fact that the rotor can be disconnected from the shafts above and below it for assembly and dismantling, thus reducing the cost of the power station building.

Fabrication by electric welding was largely adopted in the construction of the generator parts. Thus the stator frame is built up of steel segments welded together to form a structure of great strength and rigidity. It is made in four sections which are bolted together, and dovetail bars are provided on the inside to receive the core which is built up of thin laminations of high-grade iron having dovetail slots. Open type slots stamped out in the periphery of the core carry the winding

which is of the two-layer basket type composed of laminated copper with mica insulation.

The mechanical design of the rotor is based on the stresses which occur at the maximum runaway speed, with an ample margin of safety. The rotor hub (Fig. 9) is of cast steel and to this is keyed a rim formed of sheet steel laminations which are overlapped and secured by through-bolts and end plates. This ring construction, as a whole, not only gives great strength but also relieves the rotor hub of stresses otherwise imposed by the rim. The poles are built up of laminated steel ; they are provided with dovetails which fit into slots punched in the outer periphery of the laminated rim and are secured by taper side keys. The field coils are formed of copper strip wound on edge on sheet steel spools and insulated with mica.

The generator shaft is of forged steel with a forged half-coupling of the spigoted type at the lower end connected to the turbine shaft. The two half-couplings are bolted together through clearance holes and the torque is taken by radial keys.

At the upper end a steel sleeve is shrunk-on and dowelled to a collar forged integral with the shaft, the whole forming the thrust collar which transmits the load to the bearing. The rotor hub is spigoted and bolted to the top of this collar, and the torque transmitted by radial keys.

The centre of the shaft is drilled throughout its length to take the oil pressure pipes for the runner blade servo-motor. The thrust bearing (Fig. 11) designed for a total load of 750 tons, is mounted in the lower bracket and consists of pivoted steel babbit-lined segmental pads in a steel housing, or thrust pot, each pad being individually adjustable. A guide bearing is mounted in the same chamber as the thrust bearing, its bearing surface being provided by the vertical face of the thrust collar.

Doors are provided in the walls of the thrust pot between the bracket arms (Fig. 10) to give access to the thrust and guide pads for adjustment and removal.

The combined thrust and guide bearing runs in an oil bath and the oil is circulated by duplicate pumps through an outside cooler supplied with water from ring mains which also supply the generator air coolers. The ring mains receive their water supply from a tapping on the turbine spiral casing.

Brakes operated by compressed air and supported by the lower bracket are provided to bring the rotating parts to rest. Arrangements are also made to use the brakes as jacks, in which case a hand-operated high-pressure oil pump is used to apply pressure to oil in the brake cylinders.

The generator is self ventilated, air being circulated by means of fans attached to the rotor rim. The

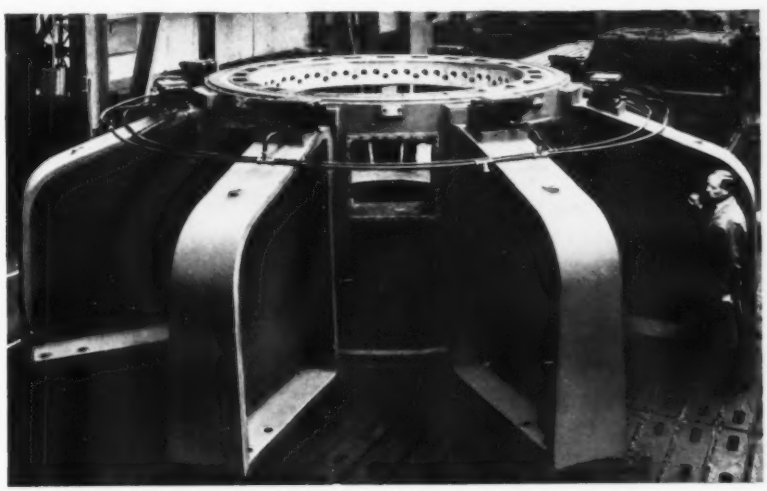


Fig. 10.

Bracket for generator thrust and guide bearing

Fig. 11. *Thrust bearing and equalising ring*

cooling is of the closed circuit type and six air coolers are arranged radially outside the stator frame.

The main and pilot exciters are mounted upon an extension shaft, and directly below them upon the same shaft is mounted the auxiliary alternator for the governor drive. The field of the latter is provided by permanent-magnet type poles.

Auxiliary Generating Set

A 500 h.p. water-turbine driven generating set (Fig. 12) is installed in the power station and supplied by a separate penstock. Its purpose is to supply the station essential auxiliary services and also some power locally at Tekapo when the main set and 110 kV transmission line are out of service. The turbine of this unit is of the horizontal reaction type, operating under a net head of 74 feet and running at 600 r.p.m. The generator is designed for a continuous output of 400 kVA, 320 kW at 0.8 power factor, 400 volts 3-phase 50 cycles. The set is provided with a belt-driven oil pressure governor of standard type.

Transformers and Switchgear

The main generator is connected through the main 11 kV switchgear to a step-up transformer bank of 11,000/110,000 volts ratio, and this is connected through oil circuit-breakers (supplied by the State Hydro-Electric Department) to the 110 kV transmission line from Tekapo which joins the existing 110 kV system at Timaru. Fig. 13

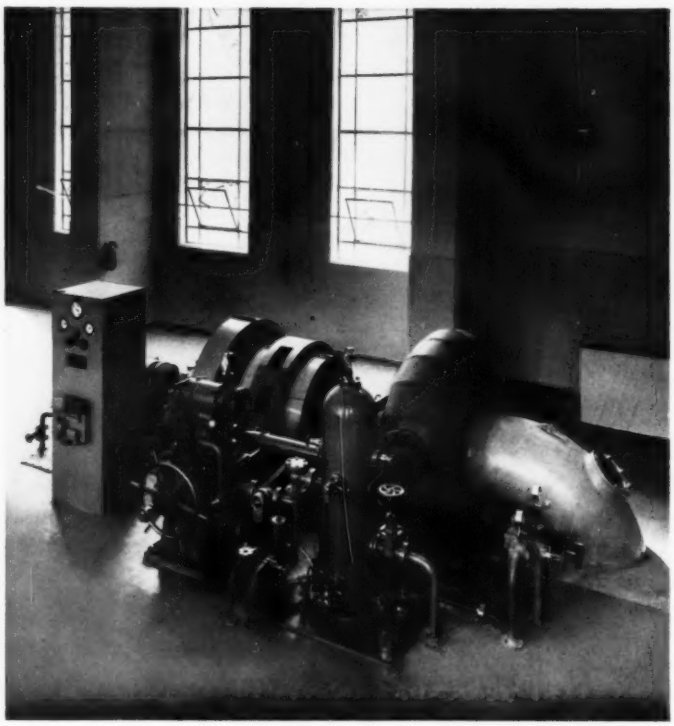


Fig. 12. *The* 500 *h.p. auxiliary water-turbine driven generating set*

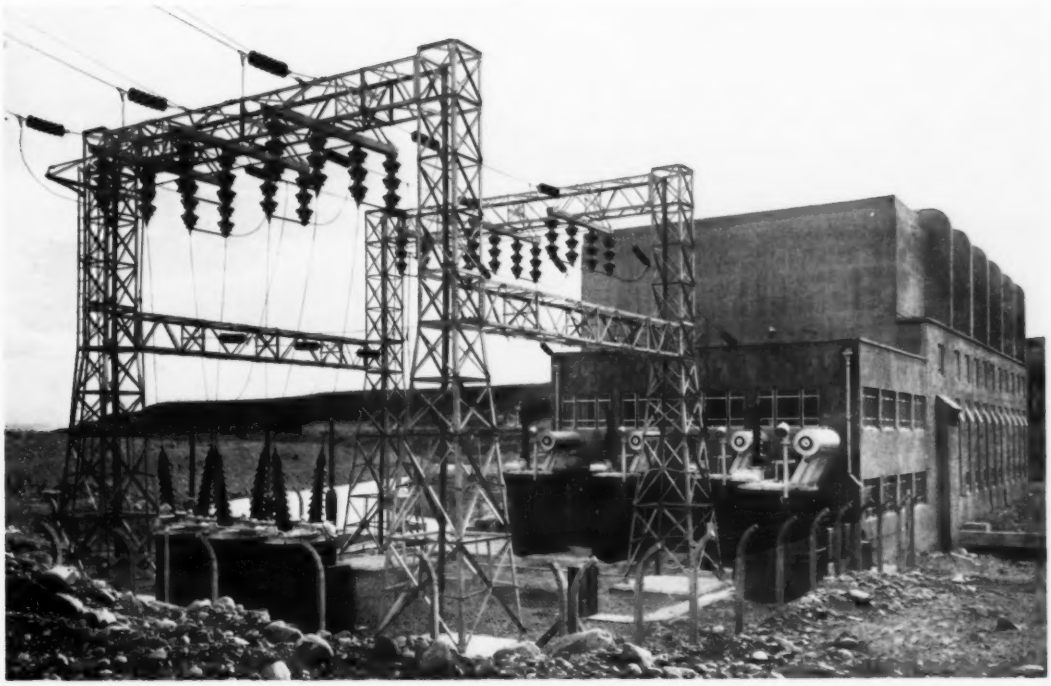


Fig. 13. *The* 110 *kV main transformers and switchgear at Tekapo connected to the transmission line to Timaru*

shows the 110 kV transformers and switchgear. A 500 kVA transformer, ratio 11,000/11,000 volts, takes power from the main 11 kV switchgear and delivers it to subsidiary 11 kV switchgear for supplying local consumers and the 300 kVA 11,000/400 volts station service transformer.

The main step-up transformer bank is connected delta-star with the high-voltage neutral solidly earthed, the 11/11 kV transformer delta-star with the neutral on the output side solidly earthed, and the service transformer delta-star with the low-voltage neutral solidly earthed.

The subsidiary 11 kV switchgear is housed in the control room, and the main 11 kV switchgear, the 11/11 kV transformer and the 11/0.4 kV transformer in an adjoining room.

The main step-up transformer bank consists of three 9,333 kVA single-phase oil immersed outdoor type transformers, with forced oil circulation water cooled, forming a 28,000 kVA three-phase unit. A fourth single-phase unit is held as a spare. Forced oil circulation with water cooling was adopted, in preference to the oil immersed water cooled type, to ensure that any leakage which might occur in the cooling system would be of oil to water, since the oil pressure exceeds that of the water.

The external coolers are duplicated, one being a standby to the other, and each is able to cool the whole 3-phase bank. Full kVA tappings are provided and are connected to an off-load tapping switch capable of operation from ground level. The terminal arrangement of each single-phase unit consists of bare shedded porcelain-clad bushings, the 110 kV bushing being of the oil filled condenser type.

The 500 kVA transformer is of the 'ON' outdoor type and the 300 kVA transformer of the 'ON' indoor type. The former is provided with on-load tap changing and the latter has an externally operated off-circuit tapping switch.

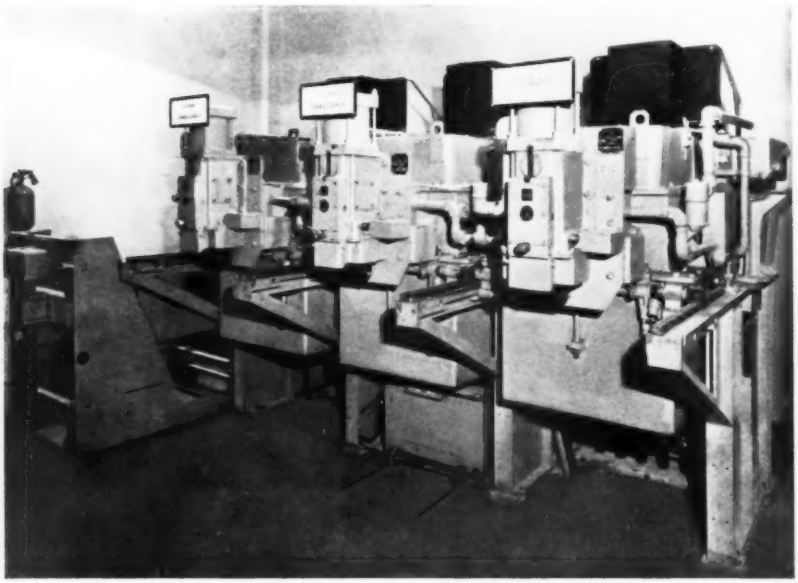


Fig. 14.
The main 11 *kV metal-clad switchgear*

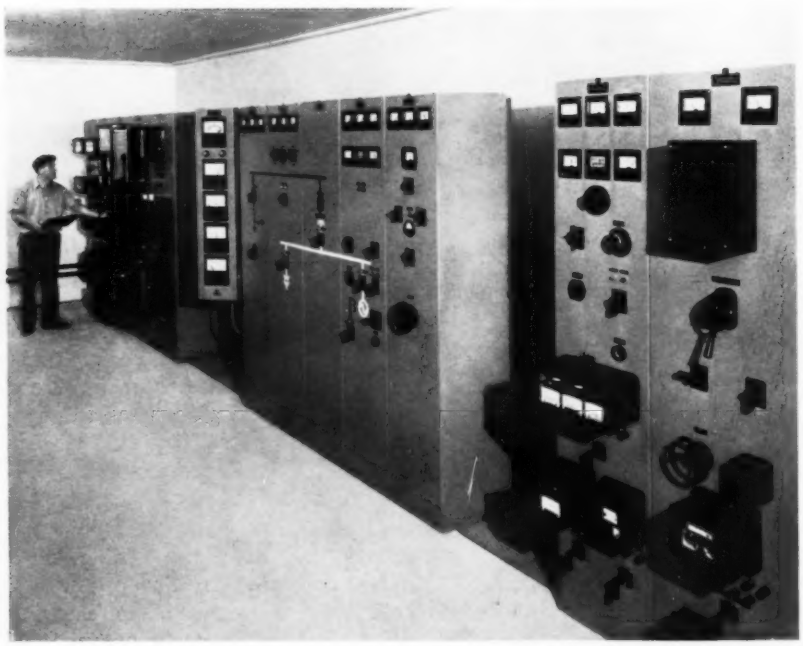


Fig. 15.
The main control board (centre) with separate boards for relays and auxiliaries

The main 11 kV switchgear (Fig. 14) has a breaking capacity of 350 MVA and is of the ' OLE 1B ' metal-clad compound filled type, arranged for single and duplicate busbar service, horizontal draw-out isolation and electrical operation. The subsidiary 11 kV switchgear is of the ' OLC 4B ' metal-clad compound filled horizontal draw-out type for single busbar service and manual operation, with a breaking capacity of 100 MVA. The 400 volts switchgear for controlling the supply to the main machine auxiliaries and other station and local services is of the air insulated air-break ' OB 3 ' type, mounted in sheet steel cubicles.

Control, Operation and Protection

As already indicated, normal operations such as starting and stopping the plant are under local manual control, and the control scheme is generally as for a fully attended station, but full protective features have been provided as for an unattended station. The load can be held at a pre-determined value by automatic load control equipment.

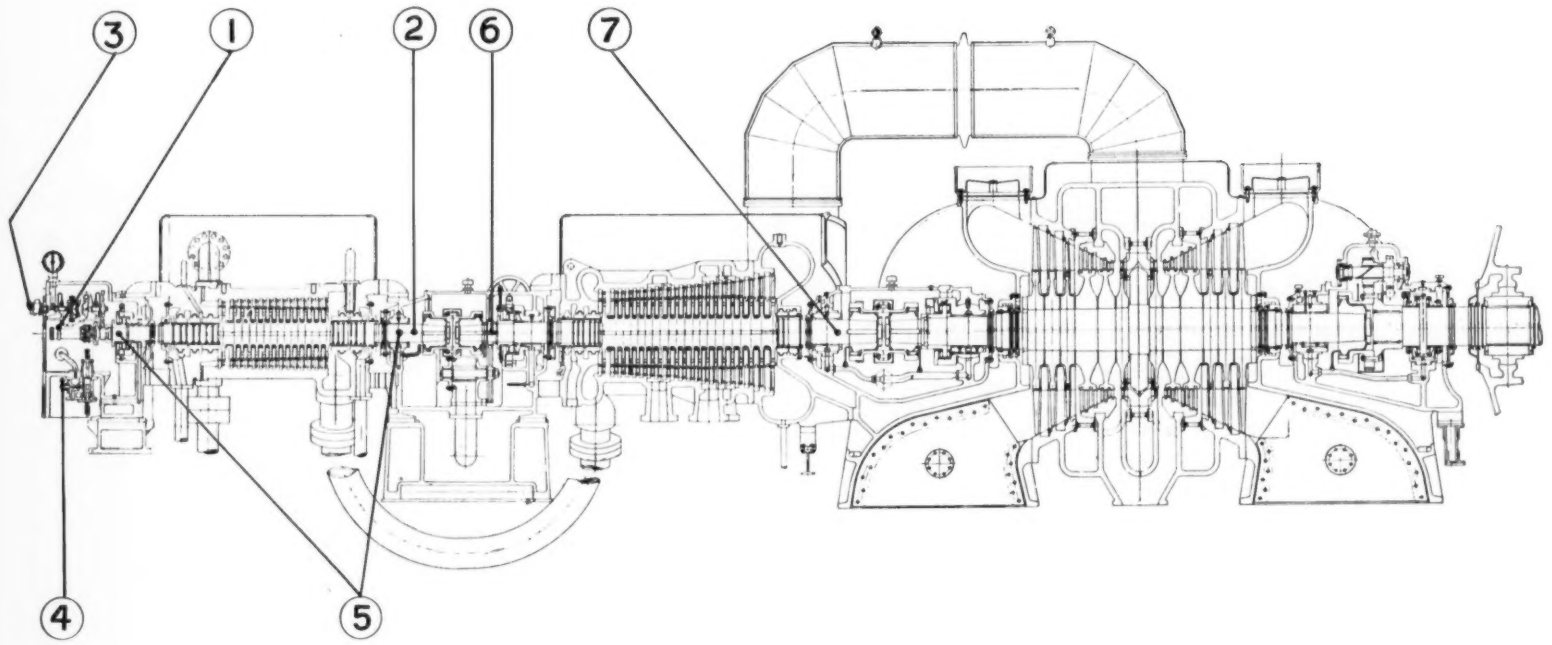
In starting the main generating unit, the operator first closes the head-gate control switch located on a panel in the control room. This causes the head-gate to open slightly and thus fills the penstock, after which the head-gate opens automatically. The auxiliary motors are then started from their control panel in the machine room, and the main generating set runs up to speed and is put under governor control. From the control room, the operator closes the field circuit-breaker, raises the generator voltage to normal and puts the automatic voltage regulator into service. The machine is then synchronised.

Shutting-down is effected by first reducing the load and opening the 11 kV generator oil circuit-breaker and then in general reversing the starting procedure, the brakes being finally applied to bring the main set to rest. The starting and stopping of the auxiliary generating set is effected in a similar manner.

The main control board (Fig. 15) for effecting these operations is of the sheet steel cubicle type with rear access doors, and is provided with a mimic diagram. The synchronising instruments are mounted on a swinging panel on the end of the control board. The control room also contains two separate boards, of the same type as the main control board, for the relays and instruments and for controlling auxiliaries, respectively.

Model

As the manufacture and successful commissioning of the Tekapo plant, representing the most modern development in water turbine and water-wheel alternator design, was recognised as an event of some importance in the field of hydro-electric engineering, the large model shown in Figs. 6 and 8 was made to the Company's order and displayed at the Festival of Britain South Bank Exhibition in London during 1951 where it created much interest amongst both engineers and the general public.



1. H.P. rotor eccentricity detectors
2. H.P. rotor position detectors
3. Tacho-generator
4. Pedestal expansion transmitter
5. Vibrators (on outside of pedestal)
6. I.P. rotor eccentricity detectors
7. I.P. rotor position detectors

Fig. 1. *Sectional arrangement of a* 3*-cylinder turbine, showing location of detectors*

Supervisory Aids in High Pressure High Temperature Steam Turbine Operation

By L. J. HIGGINS, Instrument Engineering Department,
and P. F. CARSON, Outside Erection Department.

IN CONSIDERING THE need for supervisory aids in high pressure high temperature steam turbine operation we are primarily concerned with the H.P. turbine, the principal feature which influenced the development of supervisory equipment being the considerable mass difference between cylinder and rotor in this stage.

The mass difference between H.P. rotor and casing (Fig. 2) presents the problem of positive and negative differential expansion between the two parts under transient heat conditions. The rotor is located axially by the H.P. Michel thrust assembly at the forward or steam end of the casing. When this thrust assembly is moved fully forward by means of the adjusting mechanism the internal axial clearances between fixed nozzle plate and rotor blading are in their most efficient setting, and when the thrust assembly is moved back towards the H.P. exhaust these internal clearances are increased.

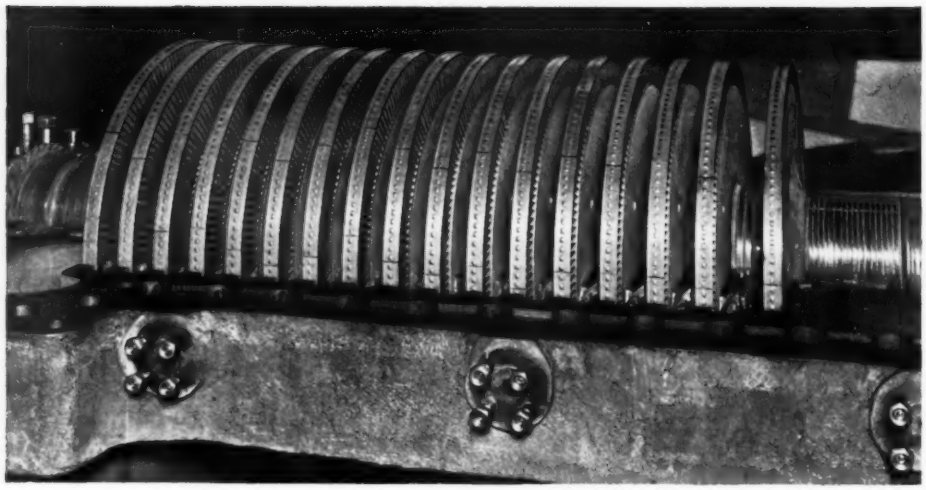
The casing is located at each end on paw grip supports, the exhaust end fixing being the intermediate pedestal in a three cylinder unit. The steam end location is the steam end pedestal. Both steam end and intermediate pedestals are designed to slide on axial keys on their respective stools, thus accommodating axial expansion of the two cylinders, high pressure and intermediate pressure. Transverse expansion of these cylinders is arranged for in the paw grip design which simultaneously compensates for vertical casing expansion.

Shaft Position

When starting a turbine of this type, the first application of heat has the effect of expanding the rotor along its length at a greater rate than the casing. The maximum positive differential (relative expansion) towards the exhaust end could be excessive without indication of such a condition and means of compensating control. Conversely, a severe reduction in heat when the unit is running under load conditions will cause a rapid negative differential—or contraction of the rotor relative to the casing—which must be counteracted to prevent axial fouling within the cylinder. It is desirable to measure differential conditions at the exhaust end of the cylinder in order to record the total expansion value of the rotor over its full length from the H.P. thrust point.

A pair of electro-magnetic detectors located in the bearing pedestal, and having their poles presented to the surface of a disc on the shaft at its exhaust end, are connected as shown diagrammatically in Fig. 3. The output from the circuit is calibrated on an indicating and recording instrument to show the position of the rotor relative to the casing. A convenient reference point is taken where the rotor is in the ' full on ' position, with the thrust hard-up, and this constitutes the mechanical zero of the circuit. The electrical zero of the instrument corresponds to a relative movement of 50 mils negative, and full-scale deflection represents a relative movement of 150 mils positive ; positive readings relate to expansion of the rotor in the direction of the steam flow from the reference point described.

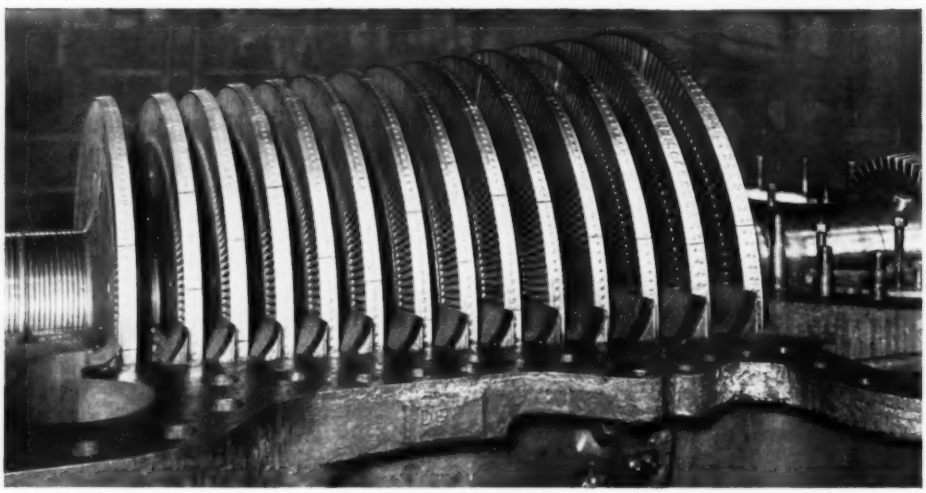
A typical record of differential expansion occurring in the H.P. cylinder of a 60 MW turbine having initial steam conditions of 900 p.s.i.g. and 900° F. is shown in Fig. 4. Further, in Figs. 5 and 6, graphical analyses of all the factors influencing differential expansion within the H.P. cylinder are shown. Fig. 5 relates to the conditions during the running up and loading period of the unit and shows the steadily rising value of positive differential in the initial stages. Fig. 6 shows the conditions applying during load rejection and the shutting down of the plant. It will be seen that a rapid increase in negative differential value occurs, and



(*a*)

Fig. 2. Two examples of an H.P. cylinder and rotor, illustrating (*a*) *the mass difference and* (*b*) *a closer relationship in mass*

(*b*)



is compensated for by the admission of superheated steam at the shaft glands. The point of admission of superheated steam is indicated. Curve B in both cases indicates adjustment to the H.P. thrust bearing axial setting to compensate for expansions and to keep the rotor axial position within the prescribed working range. The axial clearances within the machine at any time during running are obtained by subtracting corresponding values of B from curve A. This difference is given by the shaft position indicator. It is sufficient for the operator to understand that the pointer of the instrument must be kept within certain limits on the scale, and he is assisted in this by the parts of the scale outside the preferred operating range being filled-in red. Any tendency to pass these limits is counteracted by the methods just described.

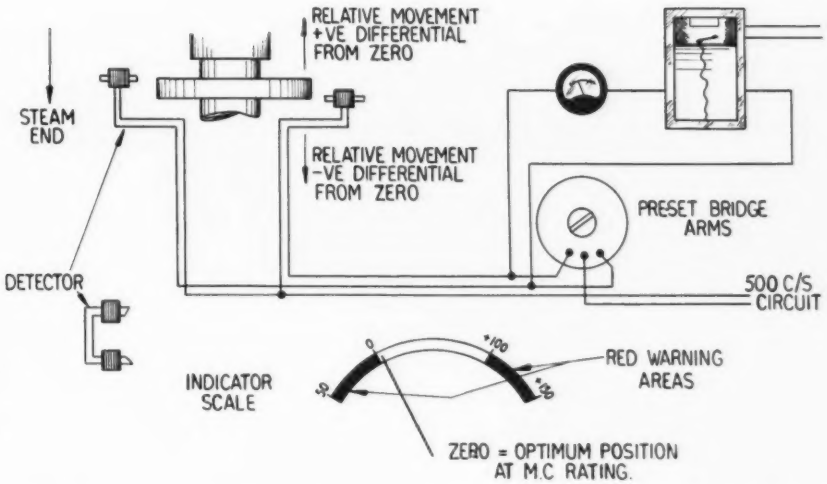
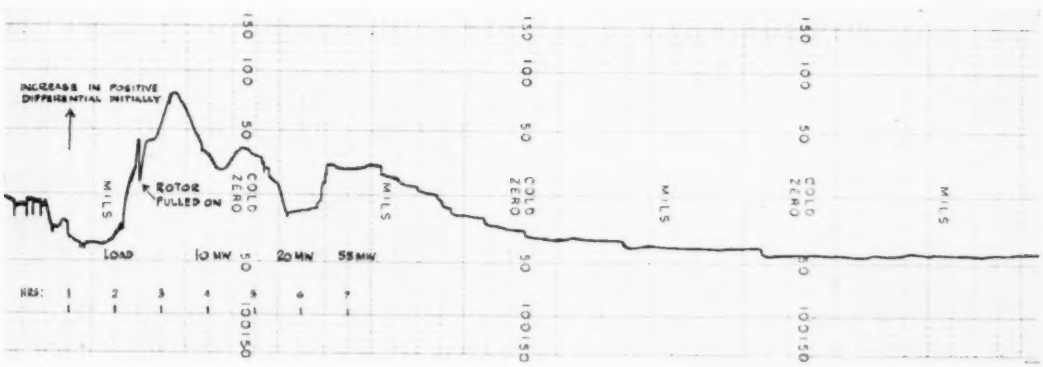


Fig. 3. The rotor position instrument

Fig. 4. (below) Typical recording of rotor position in the H.P. cylinder of a 60 MW turbine



In certain cases an optical shaft-position indicator (Fig. 7) may be fitted, wherein the magnified images of a small disc on the shaft and a gauge on the casing are projected on to a screen. The position of the shaft in relation to the casing can thus be observed and movements read by means of a scale engraved on the screen.

Eccentricity Indicator

Of the other factors affecting satisfactory operation, possibly the most important is shaft eccentricity. Application of steam to the rotor glands when starting up the machine often has the effect of causing slight deformation of the shaft, due to local heating in the vicinities of the H.P. and exhaust end glands. This deformation is indicated as eccentricity.

The radius of an eccentric circle described by the centre of an extension of the shaft at the H.P. overhang (Fig. 8) is shown on an indicating instrument. A permanent record is also provided. Electro-magnetic detectors (Fig. 9) are arranged about a disc on the shaft and the radial variation of the air-gap between their poles and the disc surface is calibrated in terms of eccentricity by means of a circuit arranged as in Fig. 10. It is true that the eccentricity at this point does not give the maximum deflection of the rotor but, since it is the degree of the condition which is required, this is not a disadvantage from the point of view of operation. Readings of eccentricity are independent of the lifting of the shaft. 'Climbing' in the bearings is usual with increase in speed or

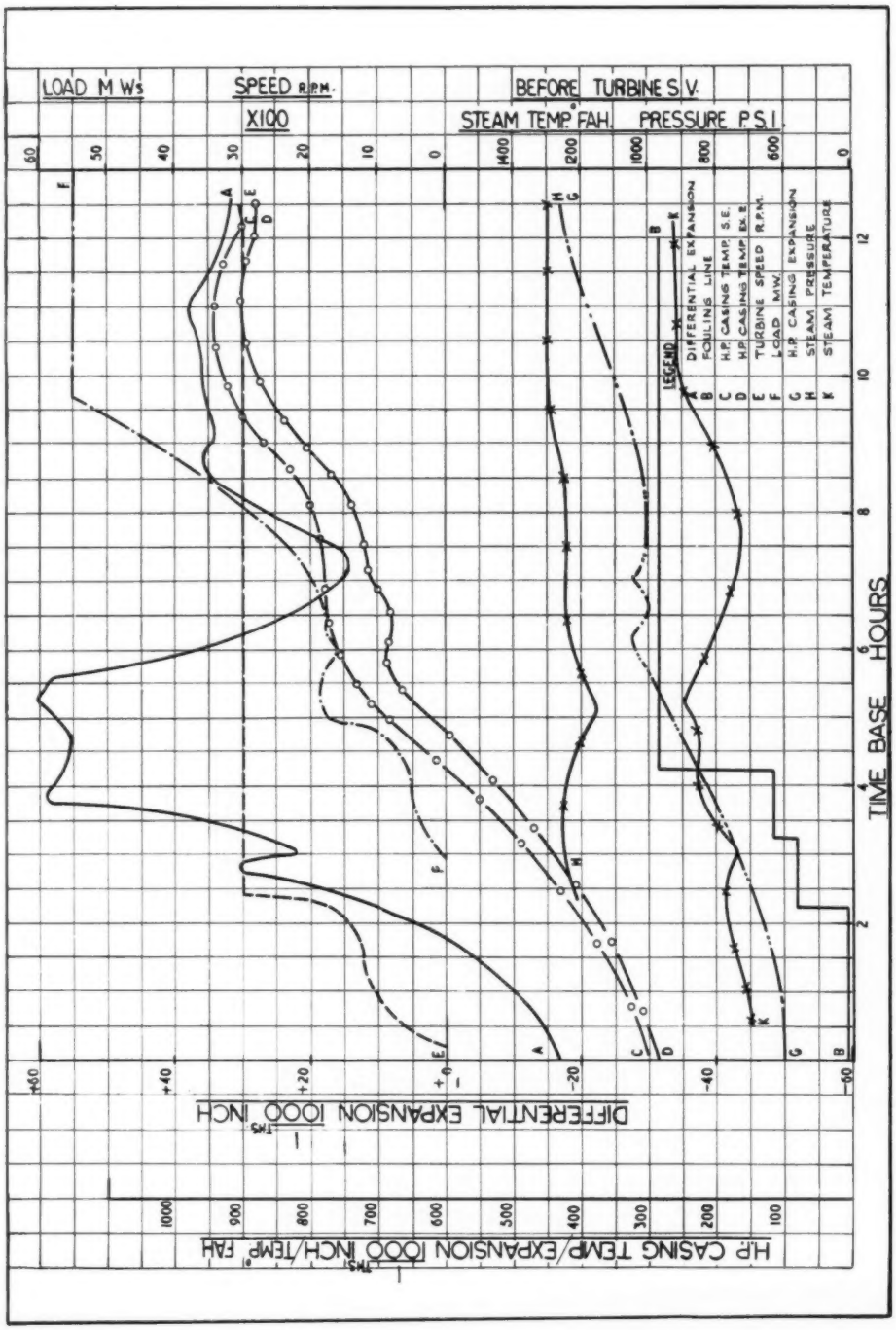


Fig. 5. Graphical analysis of factors influencing differential expansion when running-up and loading

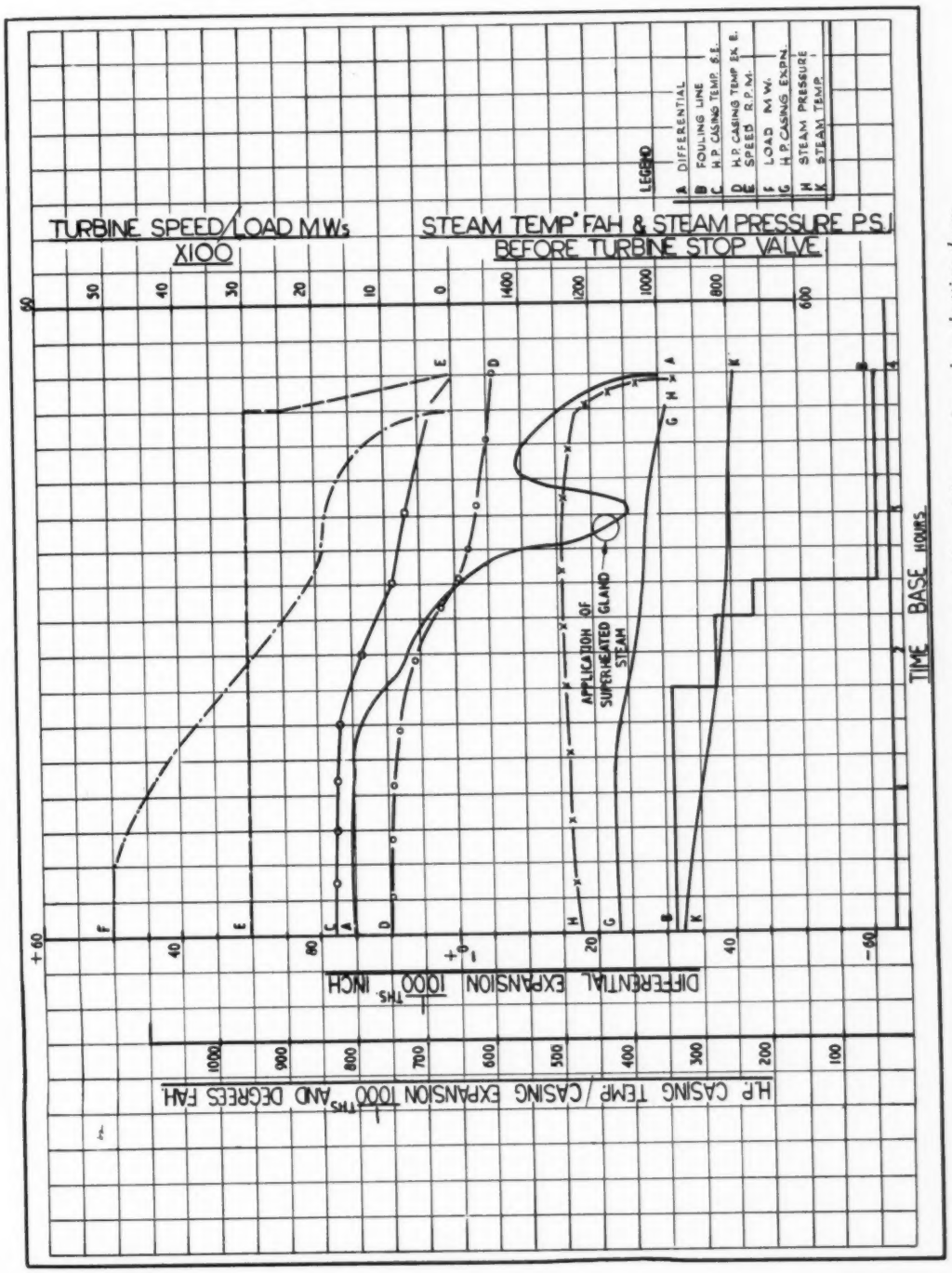

LEGEND
A DIFFERENTIAL
B FOULING LINE
C H.P. CASING TEMP. S.E.
D H.P. CASING TEMP EX. E.
E SPEED R.P.M.
F LOAD M.W.
G H.P. CASING EXPN.
H STEAM PRESSURE
K STEAM TEMP.
TURBINE SPEED/LOAD M.Ws X100
STEAM TEMP° FAH & STEAM PRESSURE P.S.I BEFORE TURBINE STOP VALVE
APPLICATION OF SUPERHEATED GLAND STEAM
TIME BASE HOURS
DIFFERENTIAL EXPANSION 1000 THS. INCH
H.P. CASING TEMP / CASING EXPANSION 1000 THS AND DEGREES FAH

Fig. 6. Graphical analysis of factors influencing differential expansion when shutting down

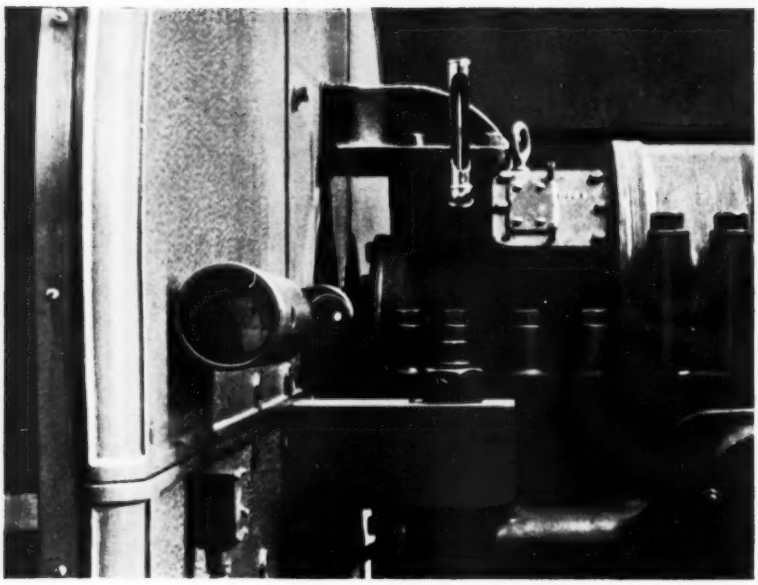


Fig. 7. The optical shaft-position indicator showing projected image on viewer screen

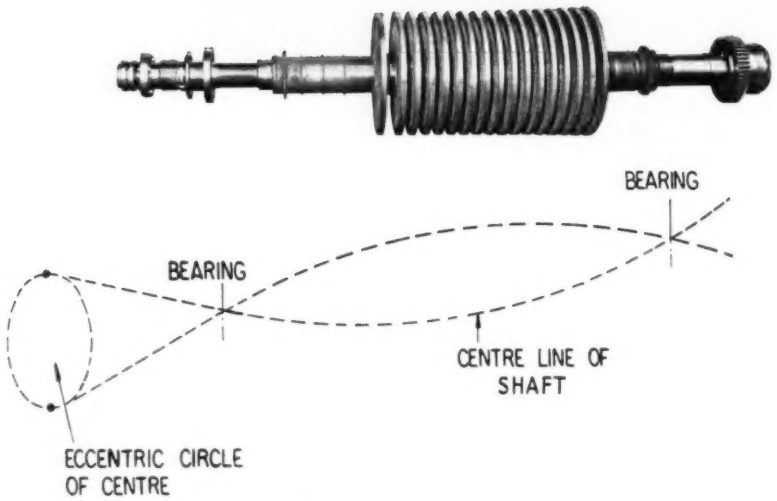


Fig. 8. H.P. rotor and diagram showing eccentric circle described by the end of a shaft overhang

change in oil conditions. Removal of the effect of these conditions on the eccentricity reading is achieved by providing a manual control whereby the reference centre can be adjusted. Once the machine is at speed, a system of relays initiated from the speed-measuring circuit automatically modifies the circuit characteristics so that manual adjustments are no longer necessary. By this means a fast time constant is retained during the running-up period and transient increases in eccentricity are recorded when passing through the range of critical speeds.

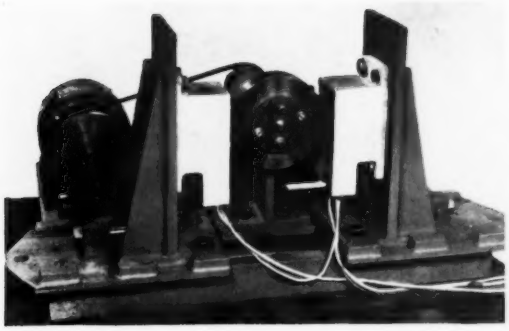


Fig. 9. Eccentricity detectors set up for testing

Observation of eccentricity is most important in evolving the operating cycle and subsequently adhering to the recommended procedure. Values of eccentricity existing initially may have a 'carry over' effect to the later stages of running, and the importance of controlled acceleration can be seen from the results obtained over the critical speed range. Fig. 11 gives a record of eccentricity taken on the H.P. turbine of a 50 MW machine. Fig. 12 is a record of H.P. rotor eccentricity when shutting down a 60 MW machine; the curve shows clearly the momentary increase in eccentricity value when passing through the critical speed range of the rotor.

In addition to the electrical instruments, a dial micrometer is operated by a spring-loaded spindle bearing on the disc at the H.P. shaft extension. Eccentricity may be read on this indicator from barring speed up to about 1,000 r.p.m. The spindle is then raised and locked clear of the shaft.

Turbine Speed

It follows from a consideration of rotor eccentricity that the speed of the set must be controlled carefully in the initial stages of running-up. An electrical tachometer is fitted which works throughout the full speed range of the machine, thus obviating the limitations of the normal mechanical tachometer which does not come into operation until speeds in excess of 900 r.p.m. are reached.

The electrical tachometer comprises an induction tacho-generator coupled to an auxiliary drive from the turbine, and a recording instrument providing a continuous record of speed and operating so that any speed variations under fault conditions, or due to some other abnormal occurrence, appear on the chart, so giving important information which would not normally be available.

Pedestal Expansion

Another instrument (Fig. 14) shows the axial displacement of the H.P. pedestal on the sliding key with respect to the position assumed when the turbine is cold. This displacement is the summation of expansion values of all cylinders which are located at the L.P. end and are free to slide towards the H.P. end when expanding.

The movement is measured by means of a lever arm, deflected by the steam end pedestal, which operates a plunger passing within two solenoids in the 'transmitter' unit. The solenoids form two arms of a bridge system which is balanced when the machine is cold. Displacement of the plunger is calibrated to give an indication and record of pedestal position in inches.

Any abnormalities are immediately shown up as breaks in the continuity of the expansion curve, and a constant check can be made to see that

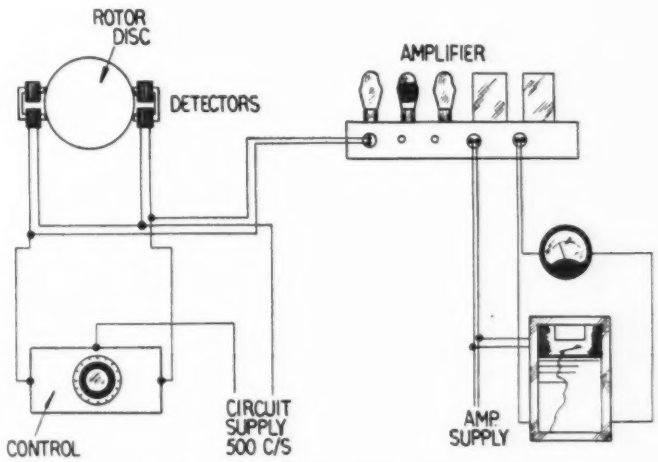


Fig. 10. The rotor eccentricity instrument

Fig. 11. Record of eccentricity taken on the H.P. turbine of a 50 MW machine

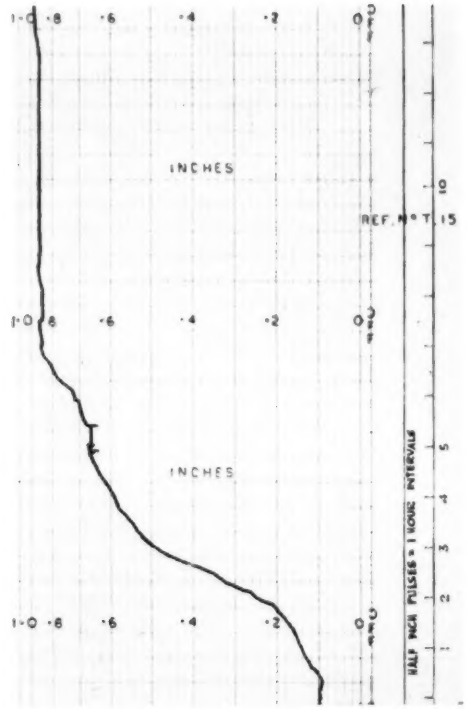


Fig. 13. Pedestal expansion record of 60 MW machine during warming-up period

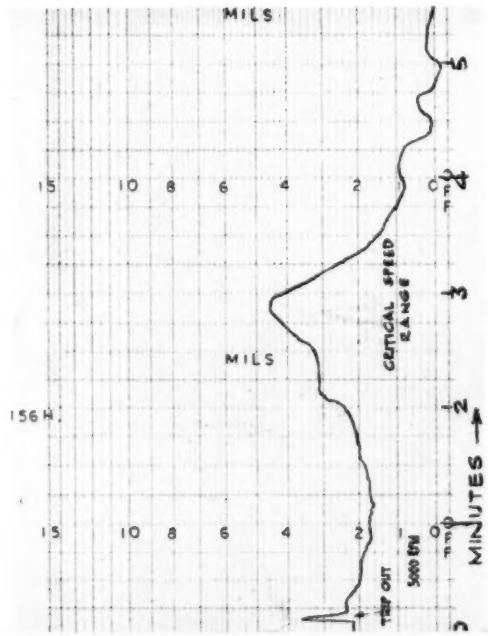


Fig. 12. Record of eccentricity showing momentary increase when passing critical range

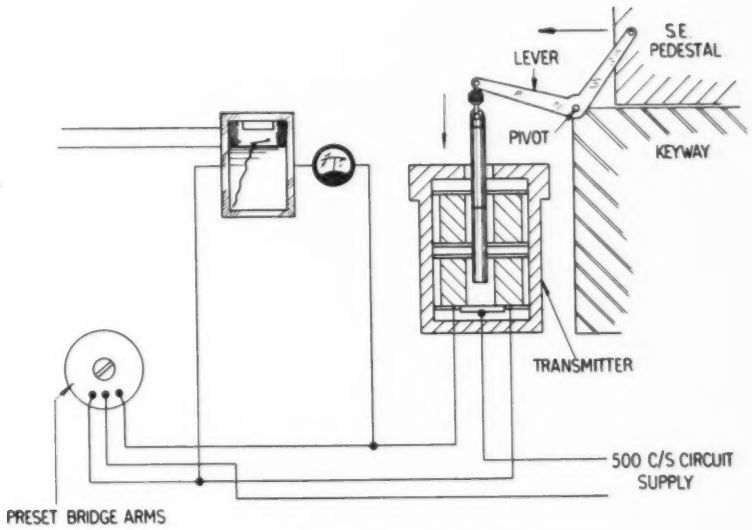


Fig. 14. *The pedestal expansion instrument*

cylinder expansions are normal. Fig. 13 shows a record taken during the warming-up period of a 60 MW machine.

This type of instrument can be adapted to show the position of the throttle valves in inches-open. Such indication is not of primary importance to the turbine operator, who has the information to hand in any case, but may be helpful where remote indicators are provided in the control room.

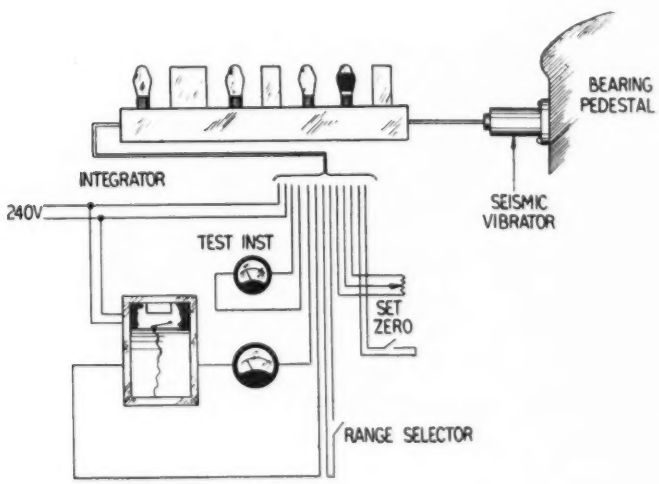


Fig. 15. *The pedestal vibration instrument*

Pedestal Vibration

In addition to the instruments already described, a guide to the smooth running of a set is provided by seismic vibrators of the electro-magnetic type (Fig. 15) which are mounted on the respective pedestals so as to produce a voltage when actuated by the transverse vibration of the pedestal. This voltage is integrated electronically and the amplitude of the vibration indicated and recorded.

Supervisory Control Panel

The indicators, recorders and associated apparatus, including the high-frequency motor-alternators supplying the measuring circuits, are housed in a supervisory control panel, typically as in Fig. 16, which is preferably located adjacent to the steam end pedestal. Recent development has resulted in a unitised system in which a basic panel providing measurement of speed, H.P. shaft

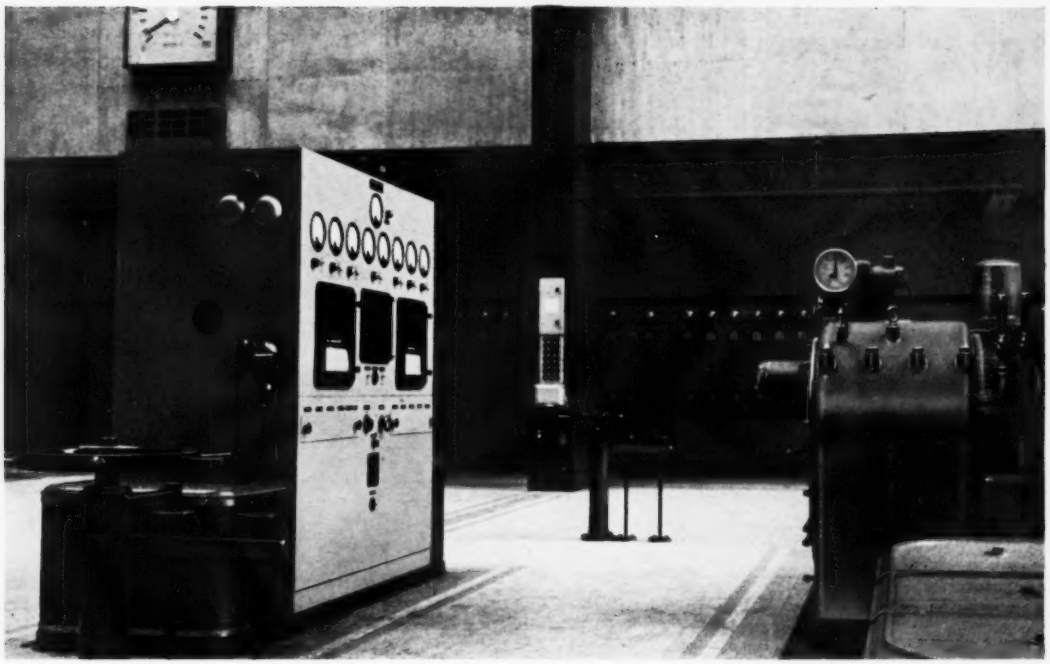


Fig. 16. *Panel adjacent to steam end pedestal, showing indicators and grouping of recording movements*

eccentricity, H.P. shaft position and H.P. pedestal expansion may be extended by the addition of other units to provide the range of instrumentation desired for any machine. In certain cases provision may be made to measure the eccentricity and position of the I.P. rotor, together with transverse vibration on any number of pedestals.

Records

The records provided play an important part in the utilisation of the instrumentation to its full extent. Charts can be operated at either of two convenient speeds which are selected by means of a switch on the panel. Operating pens working in the chart margins automatically check the points where speed changes are made and also mark suitable time intervals in the chart margin. The marking system is governed by time pulses from the station master clock. It is thus an easy matter to line up the several charts for examination and correlation of events with respect to the operating cycle. Multiple recorders are used so that records normally associated with each other are set down side by side on the same chart, *e.g.*, rotor eccentricity and speed are associated, as are shaft position and pedestal expansion, so these four records are allocated to two double recorders. A simplified chart marking system may be arranged where the station master-clock pulsing circuit is not available.

The instrumentation described, together with other instruments indicating casing-metal temperatures and steam temperatures, are essential if the running-up and loading time is to be kept to the minimum consistent with reliable operation. Careful analysis of the data obtained in first commissioning a unit, and during the early days of commercial operation, presents to the operating engineer full information on the running plant and enables the operating cycle to be determined. Departures from the recorded behaviour of the machine at any time may also enable the operating engineer to forecast potential maintenance requirements.